MECHANICAL ENGINEERING

INCLUDING THE ENGINEERING INDEX

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

THE ENGINEER IN PUBLIC LIFE

We pride ourselves in saying "Once an engineer, always an engineer." Whatever may be the course of life followed by any of us, it will always be marked by the indelible seal of the scientific, practical, and logical training to which an engineer is subjected during the early years of life.

Some have made the remark in criticism that engineers lack political intuition and ability; I would answer that a larger dose of logic and positiveness applied to politics would bring great advantage to public affairs.

Gelasio Caetani
Italian Ambassador to the United States

FEBRUARY 1923

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ADVERTISING

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Contributors and Contributions

Methods of Ash Handling

John Hunter

Methods of removing ashes from boiler rooms, and systems of conveyance are described and illustrated in an article by John Hunter and Alfred Cotton, chief engineer and chief of the research department, respectively, of the Heine Boiler Company of St. Louis. Mr. Hunter was born in Scotland. He became a seagoing engineer at the age of twenty and during the Spanish-American War was chief engineer of the *St. Paul*, operated by the U. S. Navy. In 1905 Mr. Hunter became chief engineer for the Union Electric Light and Power Co., of St. Louis, and later aided in the development of the hydroelectric station which supplies this company with 60,000 hp. over a 100,000-volt transmission line.

Alfred Cotton

Mr. Cotton received his technical education through university-extension courses and private tuition in England while serving an apprenticeship in marine engineering. In 1903 he came to America where he developed the Cotton furnace. He manufactured and installed these furnaces until the war, when he entered the employ of Colt's Patent Fire Arms Manufacturing Co.

Stresses in Electric-Railway Motor Pinions

The results of a scientific study undertaken by the General Electric Company for the development of superior electric-railway motor pinions are reported in this issue by Dr. Paul Heymans and A. L. Kimball, Jr. Dr. Heymans, who is research associate in industrial physics at M.I.T., was born in Belgium in 1895 and educated at the University of Ghent, L'Ecole Spéciale des Travaux Publics in Paris, and the University of London.

A. L. Kimball, Jr., has been research physicist at the research laboratory of the General Electric Company since 1918. He was graduated from Amherst College in 1908 and from the Harvard Engineering School in 1914. He spent the year 1919–1920 in London studying Dr. Coker's method of stress analysis by the use of transparent models.

Torsion of Crankshafts

Dr. S. Timoshenko, who has recently become consulting engineer for the Vibration Specialty Co. of Philadelphia, presents a paper which discusses how the torsional properties of a crankshaft with a single throw may be determined. Dr. Timoshenko came to the Vibration Specialty Co. from St. Petersburg, where he occupied at the Polytechnic Institute the chair in the Theory of Elasticity as Applied to Ships.

Design of Cooling Towers

Prof. C. S. Robinson of the Massachusetts Institute of Technology has established a general principle applicable to cooling-tower design and derived equations for the use of the designer. In his paper in this issue he shows by actual experiment how these formulas are applicable. Professor Robinson was formerly a chemist in the employ of the Sherwin-Williams Co., the Rosseler and Hasslacher Co., and the Walworth Manufacturing Co.

Size Selection of Dry-Vacuum Pumps

A rapid and practically accurate method for determining the size of dry-vacuum pump to employ under any set of conditions is described by Edward W. Noyes and Harold V. Sturtevant, sales engineers for the Sullivan Machinery Company of Claremont, N. H., and Chicago, Ill. Both Mr. Noyes and Mr. Sturtevant are graduates of the Massachusetts Institute of Technology, class of 1921.

Feed Heating for High Thermal Efficiency

Linn Helander, a Junior Member of the Society, presents the results of an investigation made for the purpose of determining the correct feed-water temperatures for conditions of high thermal efficiency. Mr. Helander, after his graduation from the University of Illinois in 1915, was steam and hydraulic engineer for the Pittsburgh Crucible Steel Co. During the war he was supervising engineer of tests for the Ordnance Department in the Montreal district. Since 1919 he has been with the Westinghouse Electric and Manufacturing Co.

Lumber Dry Kilns

Thomas D. Perry, vice-president and manager of the Grand Rapids Veneer Works, believes that scientific kiln drying offers engineers a splendid field for research work. His paper pays particular attention to the several classes of ventilated kilns. Mr. Perry received his A.B. degree from Doane College in 1897 and his B.S. in Mechanical Engineering from M.I.T. in 1900.

German Submarine Diesel-Engine Clutch

During the five years spent as supervising draftsman for the U. S. Navy Department, W. H. Nicholson had an opportunity to study the German types of submarine equipment. His paper gives to American builders of Diesel engines facts about German Diesel-engine clutches. Mr. Nicholson has done engineering drafting and designing for the New York Shipbuilding Corporation, the National Aniline Co., and the Newton Machine Tool Works. At present he is with the Westinghouse Electric and Manufacturing Co.

Coming A.S.M.E. Events

Pacific Coast Regional Meeting
Los Angeles, April 16-18, 1923

Spring Meeting
Montreal, May 28-31, 1923

Southern Regional Meeting
Chattanooga, October, 1923

MECHANICAL ENGINEERING

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Methods of Ash Handling

By JOHN HUNTER[1] and ALFRED COTTON,[2] ST. LOUIS, MO.

Various methods of handling ash are described and illustrated in this paper, the greater part of which deals with stationary practice, beginning with rudimentary and progressing to the most modern installations, of which schematic and actual examples are given

Methods of removing ashes from basement boiler rooms are followed by a discussion of the design, construction, and capacity of hopper ashpits, including their doors and water seals. Systems of mechanical conveyance and elevation are described, comprising ash cars, skip hoists and bucket conveyors. Fluid conveyance, as represented by water sluicing and steam-jet conveyors, is discussed in general, and typical examples of both are illustrated and described in detail. Particulars of ash bunkers and settling basins are also given.

Ash HANDLING is just as important as coal handling. It was originally accomplished entirely by hand, but with the growth of the size of boiler plants it is now either partly or entirely mechanical. There are three general methods of conveyance in use; air, water, and purely mechanical.

The great development of the central electric generating station has compelled operating engineers to give much attention and thought to ash handling. It is only a few years since boilers of 600 hp. were considered large, while today 2000-hp. units are not at all uncommon and some 3000-hp. boilers are in use. Furthermore, while boilers were usually operated at about their rating of 10 sq. ft. of heating surface to the boiler horsepower, they are now commonly operated at twice their nominal rating; and in the larger central stations it is common practice to run them for short intervals at three or even

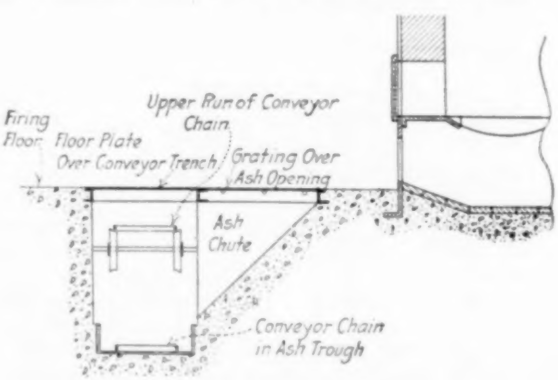

Fig. 1 Chain and Cross-Bar Conveyor

four times their rating. The bearing of this development on ash handling may be strikingly seen when we remember that a 250-hp. boiler running at rating and burning coal with 15 per cent of ash would make 150 lb. of ash per hour; while each 2000-hp. unit at 200 per cent of rating makes well over a ton of ash per hour.

While the complete paper deals with both stationary and marine practice, the present abstract is limited to the former, although definite progress in ash handling was first made on shipboard.

Except that there are several standardized types of conveyors, classification of methods of ash handling is not convenient and would serve no useful purpose. Therefore no real classification will be attempted, but several different schemes and installations will be described.

The most rudimentary method is that in which the ashes are hoed out of the fire doors and ashpits on the firing floor and shoveled into wheelbarrows. In some cases the ashes are shoveled into industrial railway cars, which are then pushed to the dumping point. The wheelbarrow can be replaced by either a mechanical or air conveyor.

Fig. 1 is a cross-section of a block-chain conveyor running in a trench under the firing floor. The ashes are hoed on to the grating through which they fall down the chute to the bottom of the trench. They are then drawn along by the chain, discharging into a bucket elevator as illustrated in Fig. 2, or on to an inclined chain conveyor carrying the ash into an elevated storage hopper.

Where there is a plentiful supply of water, a flume may be car-

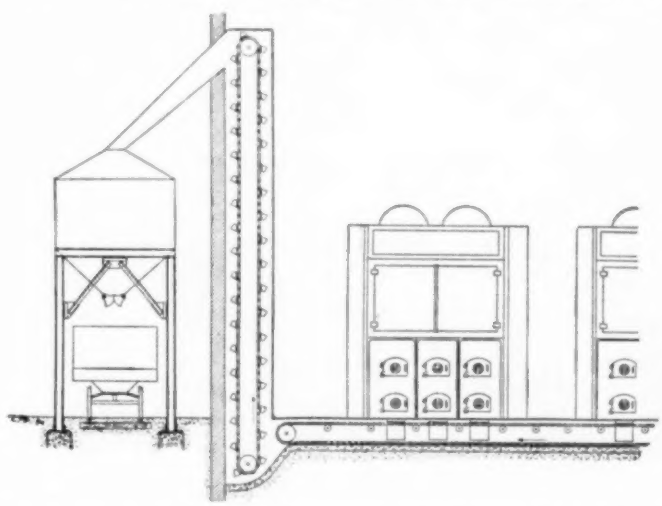

Fig. 2 Chain and Cross-Bar Conveyor with Bucket Elevator and Ash Bunker

ried along under the firing floor and the ashes raked into it through openings.

The problem is more complicated when the boiler room is considerably below the ground level, as in office buildings, hotels, etc. Whatever the final disposition may be, the ashes are invariably removed from the vicinity in motor or horse-drawn trucks. Here, the important consideration is principally that of hoisting the ashes from the boiler room to the street. In the isolated heating plants of the Union Electric Light and Power Company in St.Louis, the ash trucks are equipped with a davit just like a regular boat davit on shipboard. A small winch with a 4-in. barrel geared to a 1/4-hp. electric motor is mounted at the side of the truck. A 3/16-in. steel wire rope is attached and wound on the winch barrel, and passed through a pulley block hanging from the head of the davit with a hook at the free end. Current for the motor is conveyed by a flexible cord from a socket in the plant and snapped to the motor when the ash truck arrives. Opening the sidewalk cover, the hook is dropped and the first ash can attached, quickly hoisted to the davit head, the davit swung around and the ash can dumped into the wagon, and then returned to the basement. The whole operation is completed in a few minutes.

In some instances the air conveyor (the so-called steam-jet conveyor) has been used very advantageously in hospitals, hotels,

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and office buildings in crowded parts of cities, but it is not always applicable.

Another method is the "post box" elevator. The ashes are stored in the boiler room until the arrival of the ash truck. The elevator, which is of the bucket type, has a telescopic housing which is pushed up through the sidewalk opening and the chute extended to discharge into the truck.

The first opertion where labor can obviously be saved is in avoiding hoeing and shoveling ashes at the outset. This leads naturally to a consideration of the hopper ashpit.

Hopper Ashpits

The hopper ashpit is regular practice in the modern plant. Its the basement being deep enough for this purpose, the hopper can be quickly dumped through bottom doors. Where conveyors are installed it is usual to use a side door and work the ashes out gradually into the conveyor to avoid choking it, as would happen with a straight dump. Fig. 3 illustrates a layout of this kind. The conveyor will occupy the trench in front of the ash door. It may be mechanical, bucket or chain, or air or water, and is therefore shown schematically only.

With forward-travel underfeed stokers, chain-grate stokers, front-feed inclined stokers, and others the ashes are dumped at the rear of the furnace and the top of the hopper will be the width of the fire but of small dimension from front to rear to accord with the stoker dump. This reduces the capacity for storage as is seen by examining Fig. 5. Where large storage is imperative, the ashpit may be designed as in Fig. 4, which is drawn from an illustration appearing in *Power* of January 17, 1922. Owing to the flatness of the bottom, some hand labor is necessary to effect complete discharge when desired.

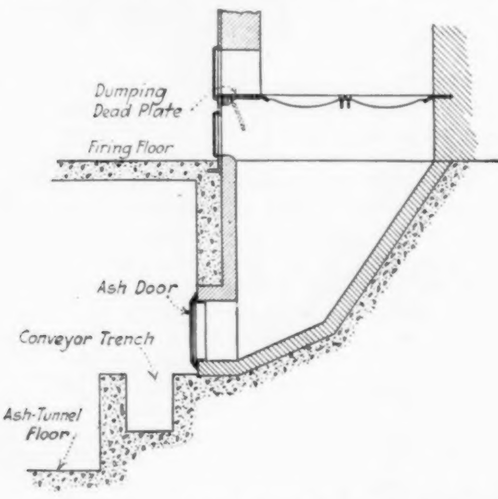

Fig. 3 Hopper Ashpit for Hand-Fired Furnace

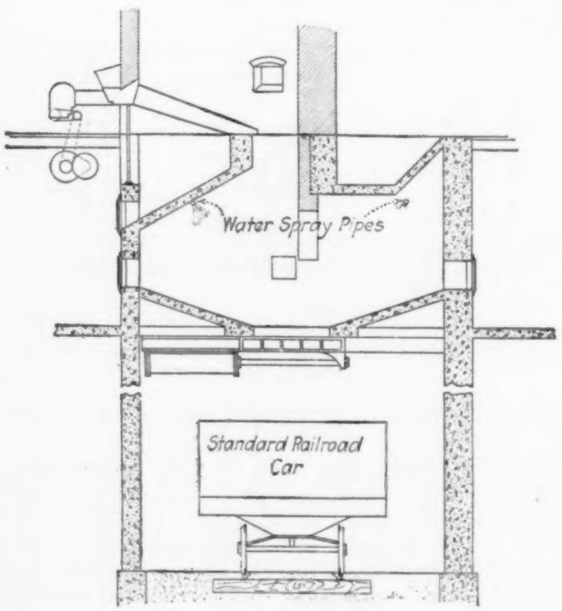

Fig. 4 Large-Capacity Hopper Ashpit

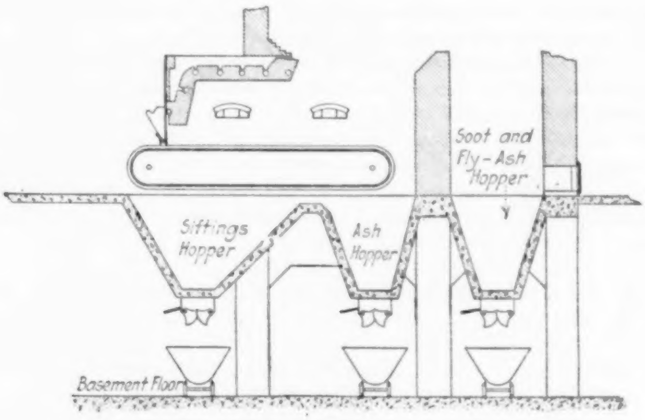

Fig. 5 Hoppers for Ashes, Coal Siftings, and Soot

With chain-grate stokers a hopper should also be provided for fine coal which shifts through the grates before ignition. Such an size and general design will depend primarily upon the manner in which the ashes flow into it, the method of removal, and also upon the system of draft. The design begins at the top to suit the stoker.

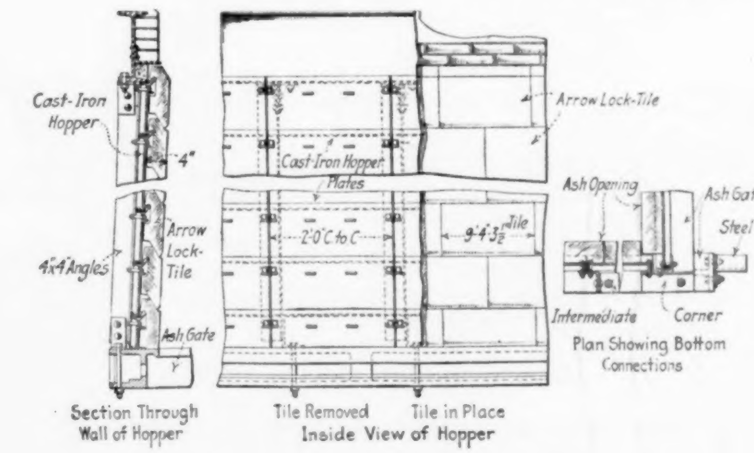

Fig. 6 Baker-Dunbar-Allen Ashpit

In hand-fired, forced-draft anthracite plants the top of the hopper will conform to the whole grate area. With stationary grates fine ash is falling constantly over the whole area. As the fires are hand-cleaned, a dumping deadplate will keep the ash away from the firing floor and allow it to fall directly into the hopper, the forced draft being temporarily shut off. With dumping grates the ash will also fall from most of the grate surface. The discharge door must be airtight to avoid increasing the cost of generating the forced draft and the possible reduction of ashpit pressure's lowering the boiler capacity. If it can be dumped directly into railroad cars arrangement is shown in Fig. 5. This coal is dumped into some means of conveyance to be returned to the stoker coal hoppers.

Side-travel underfeed stokers dump at each side of the wind box for the whole depth of the fire. Hopper ashpits for this type of stoker may therefore have greater storage capacity since the top of the hopper extends to the front of the boiler.

Capacity. In some instances very little capacity may be sufficient, owing to storage being taken care of outside the ashpit. The method of conveyance from the ashpit will have considerable bearing on the ashpit capacity necessary. But 24 hours' capacity should usually be provided in case of breakdown of conveyors, etc.

Since the required capacity depends upon the amount of ash that accumulates between emptyings, it is controlled by the rate of

firing and the percentage of ash in the coal. Multiplying the weight of coal burned between dumpings by the percentage of ash and allowing 40 lb. per cu. ft. gives the storage space required. Allowance must be made for unburned coal coming in with the ash, for neglect to empty regularly, for possible increase in the load and consequently in the rate of combustion, and for possible change to dirtier coal. The careful engineer will make the capacity of his ashpits perhaps 50 per cent greater than the calculations show, or even more.

Design. The sides should slope at not less than 45 deg. in any case, and a minimum of 50 deg. with the horizontal is preferable. If one side is vertical the opposite side may have the minimum slope; but where two opposite sides slope, neither should slope less than about 55 deg. If the slope is too small, arching of the ash is likely to occur. Where the width of the hopper would necessitate too great height to get these required slopes, the ashpit may easily be divided so as to have a number of discharge openings. An excellent example of reversed slope which results in absolutely reliable dumping is illustrated in Fig. 13.

If a very small slope is used so as to get large capacity such as in Fig. 4, access doors should be provided at the top of the slope so that the ashes may be pushed to the dump doors with a minimum of labor, and ample space should be left so that long ash tools can be wielded with ease.

Large discharge openings should be used, though their size depends to some extent upon the method of firing. Modern practice requires 30 to 36 in. as a minimum, with many instances of clear openings 5 ft. square.

The bottom discharge is undoubtedly the least costly in labor and well repays the added expense of the greater height of basement needed. The height of the bottom of the hopper above the basement floor depends upon the system of conveyance adopted. It is greatest where standard railroad cars are used, about 8 or 9 ft. to clear the cars and about 17 to 18 ft. if a locomotive must pass under. With industrial cars 5 or 6 ft. is sufficient, though a clear headroom of 6 ft. 6 in. to 7 ft. is preferable. In any case there should always be sufficient vertical space to allow a bar to be pushed up into the hopper to clear away any obstruction. In designing new plants ample headroom should always be provided.

Where storage takes place in the ashpit, water spray pipes should be provided for quenching the ashes. These pipes should be near the top and sheltered from the incoming ash. Or a substantial spray ring such as illustrated in Fig. 9 may be used. Many prefer to make the hopper sufficiently large for the ashes to remain long enough to cool naturally. If too much water is used it will leak from the dump doors and flow about the basement; and as it contains much fine ash in suspension, it will clog sewers and necessitate cleaning them frequently.

Construction. Hopper shells made of sheet steel lined with firebrick are undesirable owing to rapid corrosion from sulphur in the ashes. Shells of reinforced concrete about 6 in. thick are common and satisfactory. The most modern construction, and probably the best method so far devised, is to use a structural-steel skeleton and make the shell of substantial cast-iron flanged plates bolted together.

Owing to the heat of the ashes and the possibility of the combustion of unburned coal, the hopper shells should always be lined with firebrick, which may be of second quality. With proper quenching of the hot ashes with water sprays a lining of well-burned hard paving brick is very satisfactory.

The method of construction of the Baker-Dunbar-Allen hopper ashpit is illustrated in Fig. 6. A suspended skeleton of structural steel carries the hopper shell of heavy cast-iron flanged plates. The lining is of special firebrick blocks which will not spall under the temperature changes which occur. As seen in the left-hand view these blocks are hung from the shell and interlocked in a manner that prevents displacement but allows of easy renewal, and no mortar joints are used.

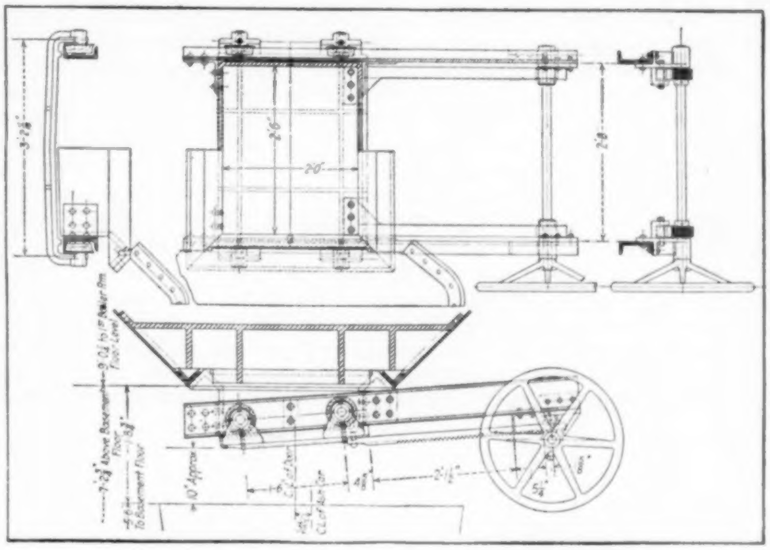

Fig. 7 Assembly of Ashpit Door and Frame

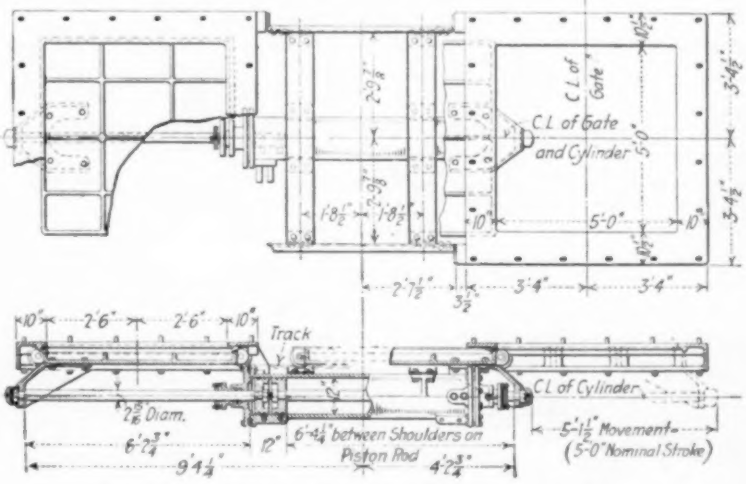

Fig. 8 Baker-Dunbar-Allen Power-Operated Ash Doors

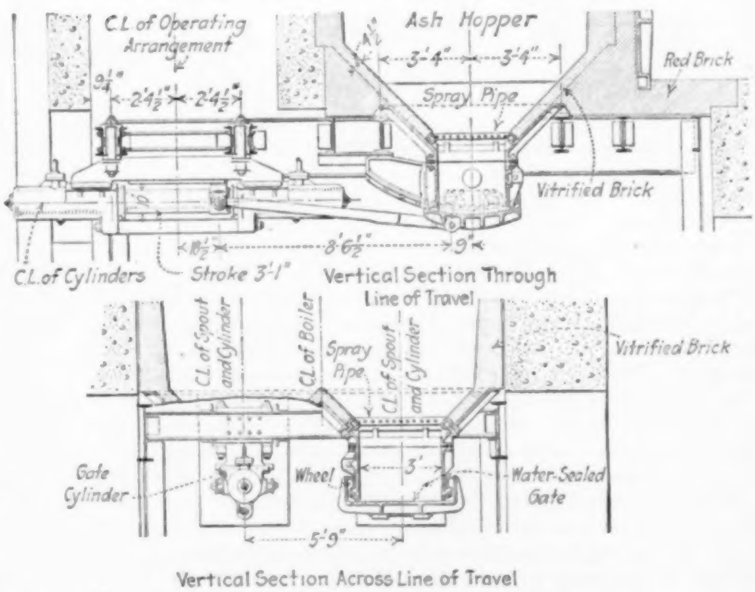

Fig. 9 Diescher Power-Operated Ash Doors

Closures. Hopper ashpits will usually be provided with doors to retain the ash and prevent the passage of air, and to allow of the ash being dumped or hoed out at intervals as desired.

Airtight ash doors are highly desirable in almost all cases. There are several ways in which the dump openings may be sealed. The

doors and faces may be machined, or the frame may be provided with a groove packed with asbestos rope, while the door has a rib or tongue which is squeezed into the asbestos packing by a crossbar and screw spanning the door. The latter method has been used with satisfaction in vertical doors like that of Fig. 3. The doors should usually be lined with firebrick to prevent warping due to hot ashes lying on them soon after dumping.

An example of ash door is illustrated in Fig. 7. These doors are of substantial construction, and are carried on rollers running on steel members. They are operated by hand with rack and pinion. Smaller doors are operated by hand, but when the openings approach 3 ft., power operation is advisable for speed. The larger hand-operated doors are worked by gearing, such as rack and pinion, while power operation may be hydraulic, compressed air, or electric.

Fig. 8 illustrates two Baker-Dunbar-Allen dumping doors arranged for compressed-air or hydraulic operation. The cylinder is worked between the openings and contains two pistons, one connected to each door. The doors are lined with firebrick.

The Diescher dumping door shown in Fig. 9 is provided with rollers which run on a curved track. Owing to the curvature of the door in conjunction with the flanges which carry the rollers, it retains water to form a seal and this water renders lining unnecessary. The door can be run off the track and another door run on if replacement is ever necessary by simply removing the pin of the driving connection. There is ample power and strength to shear easily through any clinkers which may get caught during closing.

Instead of using doors, closure may be effected by water seal. This method may be described as a U-tube with one leg forming the hopper ashpit while the other is open to the atmosphere and possibly to the entrance of a clamshell bucket. Water in the bend of the U-tube forms a seal and serves to quench the ashes.

An excellent example of this method is illustrated in Fig. 10, which is a cross-section of the boiler room of the Springdale Station of the West Penn Power Company. More than sufficient water for the seal is provided by the waste cooling water from the clinker grinders. The overflow is from one ashpit to the next, until it is finally discharged from the last ashpit. This system of ashpit requires considerable depth of basement—not less than about 30 ft. from firing floor to basement floor.

A further advantage of this method is that no combustible or corrosive gas escapes into the basement. When boilers are being pushed with heavy loads, gas often escapes from hopper dump doors. Apart from discomfort and danger to those in the basement, considerable corrosion of ironwork often occurs. Such troubles are entirely obviated with the water-seal ashpit.

Mechanical Conveyance

The method of conveying ash by emptying ashpits into small dumping cars has a great deal to recommend it. The cars are inexpensive, can be moved by men, animals, tractors, or locomotives, and can be run about the floor or on tracks, all according to the amount of ash to be moved and other conditions.

In the Ashley Street Station of the Union Electric Light and Power Company, St. Louis, Mo., a system of industrial railway and ash cars was installed in 1905. The basement floor of this plant is 30 ft. below the flood stage of the Mississippi River, and this necessitated a watertight basement with consequent elevation of ashes.

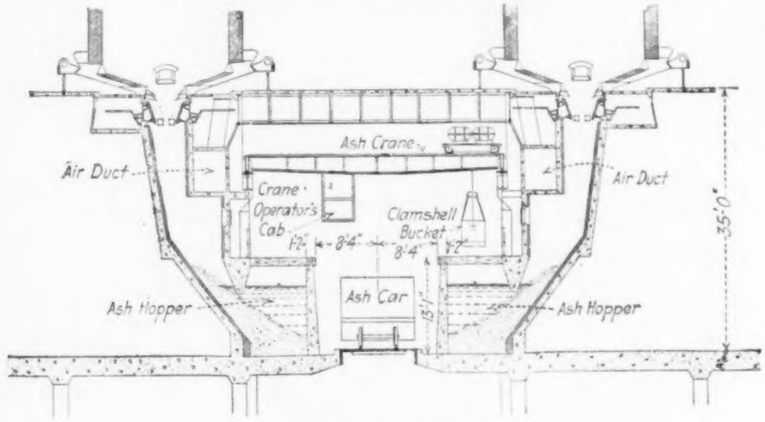

Fig. 10 Water-Sealed Ashpit

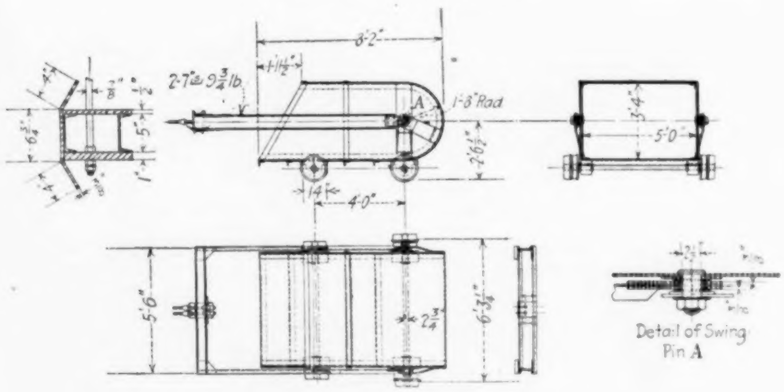

Fig. 12 Details of Skip-Hoist Bucket

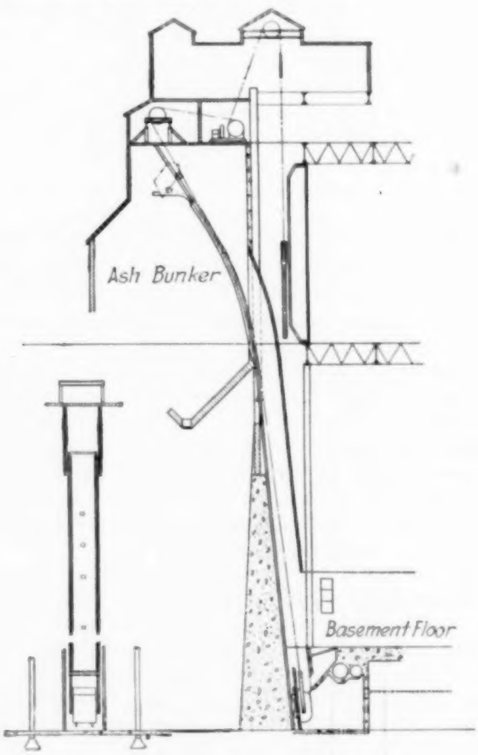

Fig. 11 Skip Hoist

The boilers are equipped with chain-grate stokers discharging ash into hoppers having a capacity of 24 hours. These hoppers are of steel plate lined with vitrified brick laid in cement, and each has two large horizontal dumping doors which are illustrated in Fig. 7.

The railway is 30 in. gage and the dumping cars have a capacity of three tons each. The cars are lifted 120 ft. to an elevated ash bunker by a skip hoist installed in 1916 to replace the car elevators originally employed; a gasoline locomotive is used to haul the ash cars. The general arrangement is illustrated in Fig. 11. The details of the bucket are presented in Fig. 12.

The operation of the skip hoist is mainly automatic. When an ash car has been dumped into the bucket, a switch button is pressed. This starts the hoisting motor, which raises the bucket to the top of its travel where it turns over and discharges into the elevated bunker. A switch trip at this point reverses the motor and thus lowers the bucket to its starting point, where another switch trip stops the motor.

The operation of this new system is entirely satisfactory and it has given good service. It has reduced the labor cost of handling ash one-third and the cost of maintenance one-half.

In the system installed at the New Bedford Gas and Electric

Light Company's plant, the ashes collected in hopper ashpits are emptied into a storage-battery truck having a dump body of 40 cu. ft. capacity. An automatically controlled skip hoist with the skip car normally in a small concrete pit just large enough to hold the car, is installed about 39 ft. outside the boiler room. The truckman pushes a button switch which starts the elevator motor, and this raises the car, dumps it into the ash bunker, and returns it to the concrete pit ready for another load. Two men handle the ashes from the station, which has a load of about 38,000 kw. at the present time.

Fig. 13 illustrates a hopper ashpit arranged for dumping directly into standard gondola railroad cars which form part of the equipment of the station. The hoppers are lined with brick, not shown, and have a capacity of about 2500 lb. of ashes. The stokers are equipped with clinker grinders. The ash gates are of the sliding type and are operated by compressed-air cylinders. Water spray pipes are provided near the top of the hoppers for wetting down the ashes before dumping.

The present consumption of this station is 413,000 tons of coal per year. The yearly ashes from the station amount to 52,800 tons.

The ash cars are handled by an electric locomotive and are dumped into an outside pit, from which they are recovered by a crane and grab bucket and loaded into the purchaser's trucks. The proceeds very nearly pay the labor cost of handling the ashes.

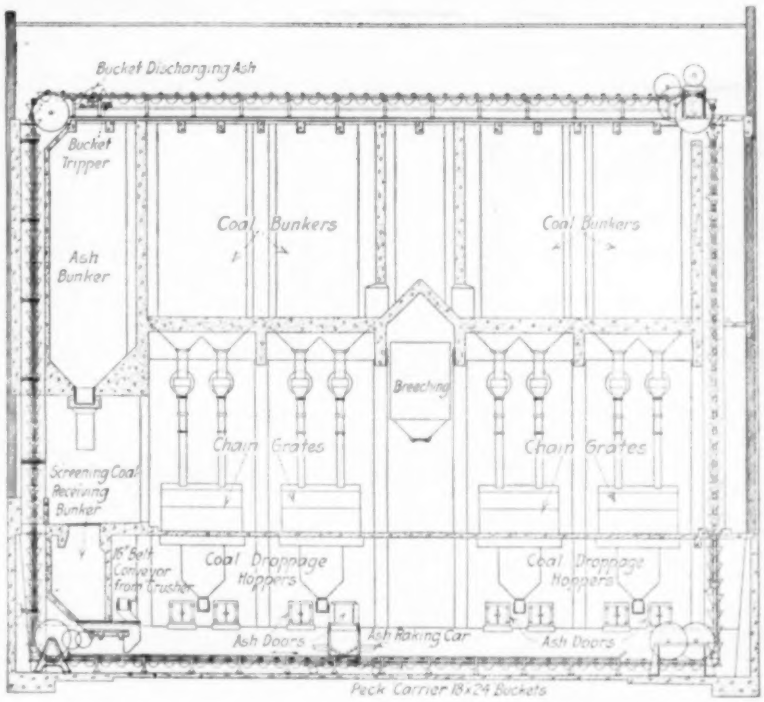

FIG. 14 PECK CARRIER RING SYSTEM

It is estimated that 45 lb. weight of compressed air is required per day per boiler to operate the hopper-ashpit dumping doors. From recent tests it is found that 1030 gal. of water is used in spraying one ton of ashes, and this amounts to about 4 per cent of the total general-service water used by the station.

Bucket conveyors such as the Peck carrier have been extensively used. With the chain and buckets forming a ring system, the conveyor is often used for coal in the daytime and ash at night.

Fig. 14 illustrates a Pack carrier installation by the Link-Belt Company in a downtown heating station where coal and ash have be handled by motor truck. The building occupies the entire site, and no projections of any kind are possible. The basement floor is 12 ft. below the alley level, and as the alley is only 15 ft. wide it was necessary to build a recess where the trucks could drive in and dump coal, and then load up with ashes.

The conveyor is located centrally between the boilers and handles coal as well as ashes, and also the siftings from the chain-grate stokers. The run-of-mine coal is screened as it is dumped from the truck, the screenings going directly through the feeder to the conveyor, and the lumps passing through the crusher. The vertical lift is 66 ft. and the horizontal centers are 85 ft. The overlapping buckets are 18 in. wide and 24 in. on centers. A 7.5-hp. motor drives the conveyor at a speed of 45 ft. per min. The buckets are discharged into the overhead bunkers by a regular tripper on the upper run.

The coal bunker is continuous along the boilers. The ash bunker which has a capacity of 130 tons, is located in line with and immediately adjacent to the coal bunker and at the same elevation. Consequently the buckets discharge coal into the coal bunker or ash into the ash bunker according to the position in which the tripper is placed. Also, the ash-bunker discharge is immediately above the recess into which the trucks are driven, just as the coal-receiving hopper is immediately below it. Therefore the trucks, which have automatic dump bodies, drive into the recess over the coal-receiving hopper, discharge their load, and receive ashes without changing their position. Quick unloading and reloading therefore materially increases the road time of the trucks; and this is a considerable item in the downtown district where traffic is more or less restricted.

FIG. 13 HOPPER ASHPIT FOR DUMPING INTO STANDARD GONDOLA RAILROAD CARS

A novel method is employed for the removal of the ashes from the ash hoppers to the conveyors. A double raking apron is contrived as a car mounted on tracks and straddling the conveyor. It is movable the entire length of the boiler house and can be brought in front of each ash door, forming a continuous chute from the hopper door to the conveyor. The operator stands on one side of the car and pulls the ashes down into the chute. After a flow of ashes has been created, water under pressure is used to keep up the flow, and the ashes are removed in a very short time. The cross-section of the plant shown in Fig. 15 illustrates this clearly.

Owing to the abrasive nature of ash the maintenance cost of mechanical conveyors is high. The ashes grind away the connecting pins, and even with regular renewals the pins sometimes wear excessively and cause breakdowns.

The life of these conveyors is considered to be about seven or eight years, and extensive repairs must be made every two or three years. The excessive cost of maintenance has led to their replacement in some installations with electrically operated cars carrying the ashes to outside pits from which they are loaded into railroad cars with a bridge crane, and the cost of handling has been reduced by 50 per cent.

Hopper ashpits with large doors should not be dumped directly on to chain or bucket conveyors. With direct

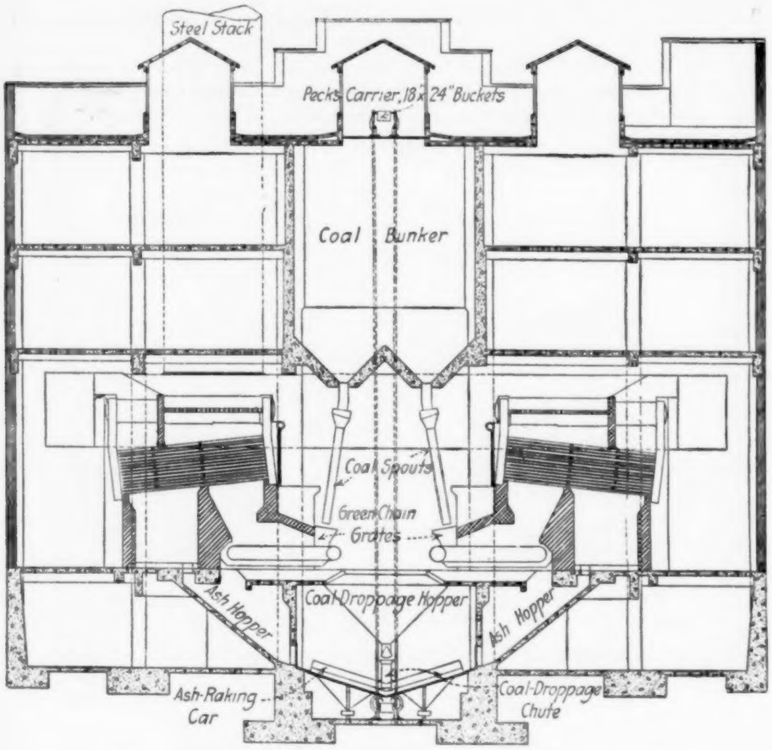

Fig. 15 Cross-Section of Boiler Room with Peck Carrier

dumping, large clinkers are liable to jam and cause breakdown, and opportunity should be given for breaking them.

The Fisk Street and Quarry Street stations of the Commonwealth Edison Company of Chicago are each equipped with bucket conveyors which handle both coal and ashes. There is an additional pair of ash conveyors at Quarry Street which carry the ashes to the train shed. The electrically operated ash cars now used in place of the apron conveyors formerly employed are dumped into a pit outside the building. A bridge crane takes the ashes from the pit with a grab bucket and loads them into railroad cars. The cost of maintenance as compared with the apron conveyors is almost negligible. The outside pit and crane were originally installed for and used with the apron conveyor.

At Northwest and Calumet stations, Chicago, the ash is dumped from hopper ashpits directly into railroad cars. The only maintenance required is upkeep of ashpit linings and dumping doors. The cost of the ash-handling system at these stations cannot well be estimated because there is none.

An example of a simple chain and cross-bar conveyor is illustrated in Figs. 1 and 2. The conveyor installed at one of the municipal plants of the Poplar Borough Council in England and illustrated in Fig. 16, may be considered as a development of this idea in combination with the water seal of Fig. 10, though there is no ashpit storage. In fact, there is no ashpit, only sheet-steel unlined chutes ending in cast-iron nozzles which dip below the water level in the conveyor trough, thus providing the water seal. The return of the chain is also below the surface of the water. The transverse centers of the chain are $19^1/_2$ in. apart and the cross-bars are $25^1/_2$ in. pitch. Therefore large clinkers easily fall between the upper run of the chains and cross-bars to the bottom of the trough. The end of the trough is carried upward at an angle of 41 deg. to its discharge into the ash bin, from which the ash is recovered by grab bucket.

No ash crushers are necessary as the space between chains and cross-bars is sufficient to allow large clinkers to fall through the upper run of the chain into the water; and the hot clinkers break up on falling into the water, the average size as delivered from the conveyor being about equal to that of a pea.

The length of this conveyor is about 140 ft. and it is designed for a chain speed of 20 ft. per min. One conveyor running at 9.5 ft. per min. and handling 3.25 tons of ashes per hour takes a little under 2 hp.

Air Conveyors

Air is passed through a pipe at a sufficiently high velocity to carry the ashes along with it. The air is admitted at one end and ash intakes are provided wherever required. There are two systems of generating the air current. In one, the pipe outlet is connected to an ash-storage tank in which a vacuum is caused by means of a steam-jet or mechanical exhauster. In the other, the air current is induced by a steam jet between the ash intakes and the outlet.

A typical layout of an air conveyor with vacuum storage tank is illustrated in Fig. 17. A steam-jet exhauster is attached to the top of the tank and may discharge into the atmosphere or into the chimney or a silencer to reduce the noise. It is claimed that ordinary gray-iron piping may be used, and a test is offered wherein a piece of conveyor pipe was replaced with light wrought-iron pipe which showed but little wear after a year's service carrying 5000 tons of ash.

An ash tank is not required as a part of the system. It is usual for the conveyor to discharge into an elevated ash tank supported on columns, so that carts, motor trucks, or railroad cars may be run underneath the tank and be filled quickly by opening the ash valve in the bottom.

The ash particles may momentarily attain velocities approximating to that of the steam jet at a little distance from the muzzle of the nozzle, and therefore local abrasion may be considerable. It is usual to locate the motor jet at an elbow, as it is then convenient to aim the jet in the new direction. When the conveyor pipe is very long, extra nozzles are installed in some cases. When the nozzles are arranged in straight pipe they are set at an angle to the axis of the pipe and are then not so efficient as they would be if coaxial. An air inlet is provided at the beginning of the pipe, for it must be remembered

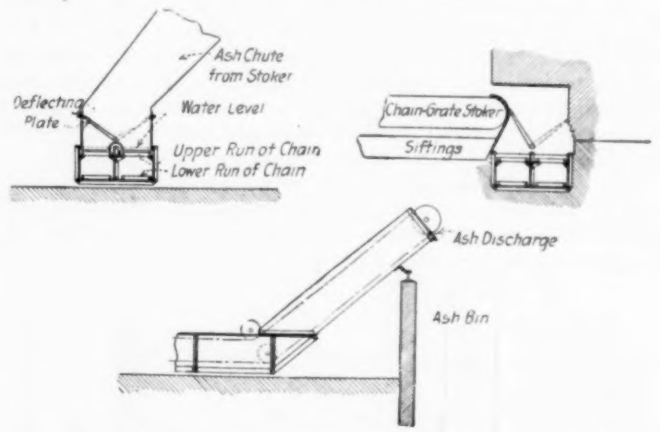

Fig. 16 Water-Seal Chain and Cross-Bar Conveyor at Poplar, England

that it is the velocity of a large body of air which carries the ash, and not the vacuum.

As an instance of the speed at which ashes can be handled, a plant may be cited containing fifteen 500-hp. boilers equipped with Murphy stokers, where the average amount of ashes in each pit ranges from 1000 to 1300 lb. One man cleaned out eight ashpits in 27 min. from the time steam was turned on until it was turned off.

Some stokers are provided with clinker crushers, and when this is not the case it is becoming increasingly necessary to add clinker crushers as part of the ash-handling equipment, so that the clinkers may be reduced to such size as the conveyors can handle without choking or breakdown.

The maximum capacity of a 6-in., conveyor is about four tons of ash per hour; that of the 8-in., six to nine tons; and of a 9-in. conveyor, ten to fifteen tons and even twenty tons in some cases. The capacity depends largely upon the size of the pieces. Ash should not be wet or quenched when fed to an air conveyor. Ashes can be conveyed

by air conveyors through a horizontal distance of about 500 ft. and through a rise of about 100 ft.

The cast iron used for pipes and elbows and other fittings is generally made of the hardest possible white iron, such as is not machinable and can only be ground, so that connections are commonly made by means of bolts slid into open lugs. The wear is greatest at elbows where the ash direction is changed, and it is usual to provide "wearing backs" of easily replaceable blocks.

Air conveyors use a large quantity of steam while running; but as they remove the ash very rapidly, the cost of steam per ton of ash removed is quite small when they are properly operated.

Considerable expense ensues if steam nozzles are allowed to wear excessively before renewal. Not only is the steam consumption greatly augmented, but the increased energy of the larger steam jet results in higher air and ash velocities with greater wear of pipes, elbows, targets, etc. Such wear of steam nozzles is greatest with wet steam.

In some instances the steam-jet conveyor is objectionable because the steam used represents so much water lost, and which must be replaced by "make-up." It is becoming increasingly general practice to distill all make-up water so as to prevent any scale-forming or foam-making salts from getting into the boilers. Condenser leakage is diligently looked for and eliminated. Therefore, owing to the care with which pure water must be conserved, it is obvious that any apparatus which removes water from the system, such as steam jets whose steam does not eventually reach the condenser, will usually be frowned upon by operating engineers.

One ton of steam will move four to eight tons of ash. With a coal containing 12 per cent of ash, two tons of steam would be used per 100 tons of coal. Taking an average evaporation of 9 lb. of water per pound of coal, the conveyor would use two tons of steam out of each 900 tons generated or nearly 0.2 per cent. To allow for careless operation or negligent maintenance and other contingencies, it would perhaps be advisable to allow, say, 0.3 or 0.4 per cent in arranging for extra distillation for make-up.

Air conveyors generally result in clean basements or firing floors because there is less spillage. The first cost is usually lower than that of a mechanical system, they take up very little space and can be installed in awkward positions, and require very little attention; but this is not always a good feature unless periodical inspection is conscientious, because steam nozzles may wear and leaks due to abrasion of pipes develop, and these may result in considerable waste of steam. The very convenience of the method may develop carelessness in allowing the steam to blow when ash is not being fed. It is safe to operate because there are no moving parts and this feature makes also for small expense of putting in renewals.

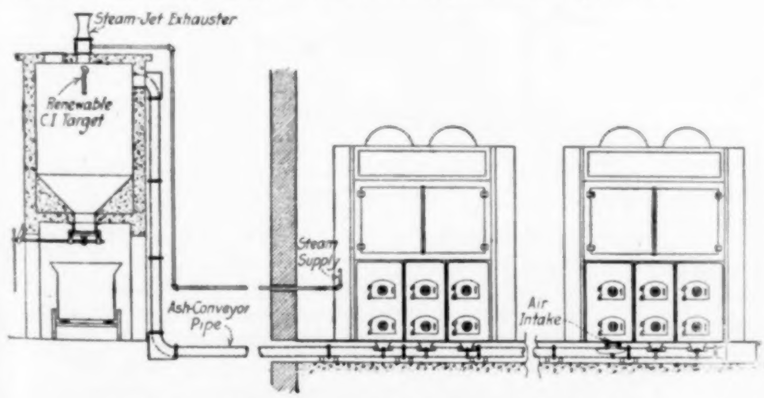

FIG. 17 AIR CONVEYOR WITH VACUUM STORAGE TANK

The air conveyors which are being installed by the Conveyor Corporation of America at the New Milwaukee sewage plant are illustrated in Figs. 18 and 19. The plan in Fig. 18 shows the ash conveyor, which has a bore of 9 in., in conjunction with the four 734-hp. stoker-fired boilers. The hopper ashpits are dumped into auxiliary hoppers formed about the conveyor intakes, as will be seen in Fig. 19. This permits of rapid feeding of the conveyor without smothering it. It is to convey a minimum of 12 tons per hour, with a steam consumption not exceeding 325 lb. of steam per ton of ashes.

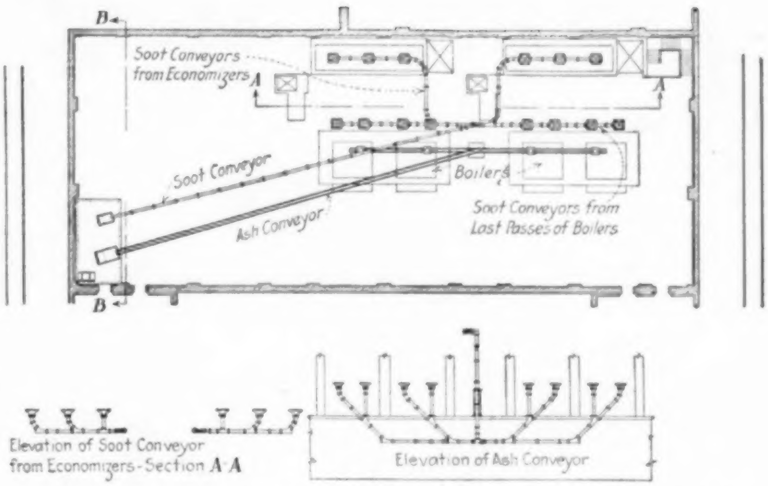

FIG. 18 AIR CONVEYOR AT MILWAUKEE SEWAGE-DISPOSAL POWER PLANT. PLAN

The conveyor for soot and fly ash is independent of the ash conveyor and has a bore of 6 in. This arrangement usually increases both the capacity and the efficiency over that obtained with combination conveyors. It is connected to the later boiler passes and to the economizer, and discharges into a separate target box set upon the top of the ash bunker.

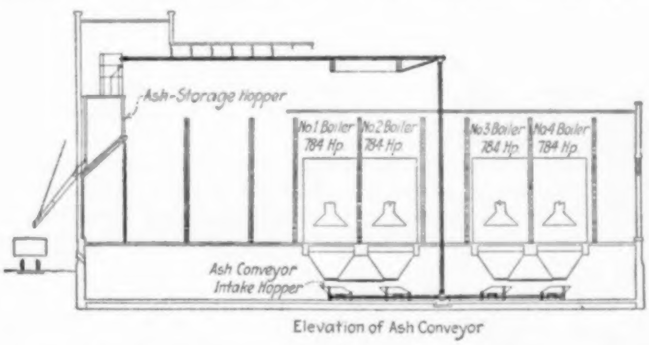

FIG. 19 AIR CONVEYOR AT MILWAUKEE SEWAGE-DISPOSAL PLANT. ELEVATION

The installation at the Lakeside plant of the Milwaukee Electric Railway and Light Company is of considerable interest because the boilers are fired with pulverized coal. See Fig. 20. There are eight boilers of 1333 hp. each—1306 hp. in the boiler proper and 27 hp. in the form of a water screen at the bottom of the combustion chamber. This water screen entirely prevents the formation of clinker or slag which was so troublesome in some of the earlier powdered-coal installations. Instead, the ash from the combustion chamber is a fine powder which is very easily handled by an air conveyor.

The air conveyors were installed by the Vacuum Ash and Soot Conveyor Company, have been in successful operation nearly two years, and practically no maintenance work has been required during that time; only one nozzle has been replaced.

To recover some of the fine ash which would otherwise be discharged from the chimney top, a smoke washer is installed in the main flue about twenty feet before it enters the base of the stack. It consists of two 4-in. pipes placed parallel to each other in the top of the flue. These pipes are provided with nozzles of $^1/_4$-in. pipe 1 in. long placed at 3-in. centers along each pipe, the nozzles of one pipe being staggered relatively to those of the other. The pipes are supplied with about 350 gal. of water per min. at a pressure of about 6 lb. per sq. in. The jets from the nozzles form a water curtain through which the gases must pass before entering the chimney.

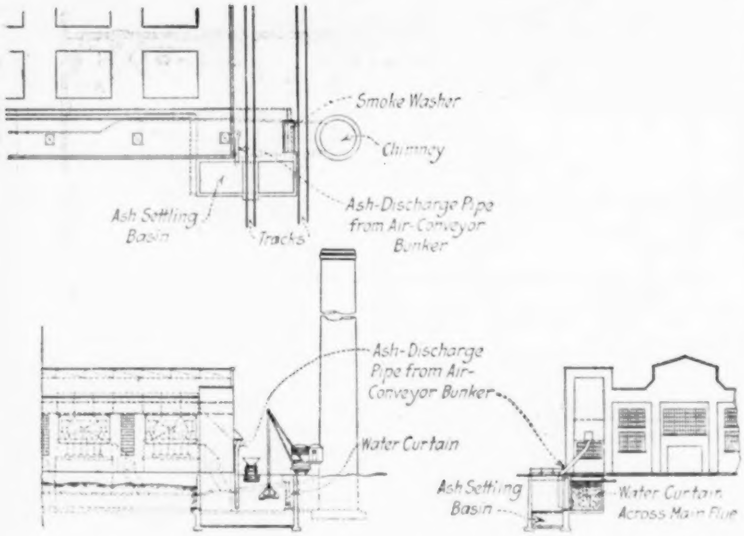

Fig. 20 Air Conveyor and Smoke Washer for Pulverized-Coal Firing at Lakeside Station, Milwaukee

Actual tests show that 50 per cent of the ash suspended in the flue gases is removed by this apparatus.

The boilers are arranged in two rows of four each, facing each other across the firing aisle. There are two conveyors, the main line of each being in the floor of the ash alley in front of the furnace ashpits. Each main line takes care of four boilers. Branch lines lead to the ash chambers at the rear of the boilers and also to the ashpits under the economizers.

All the conveyor lines are of 8-in. cast-iron pipe. The running length of the two main lines is approximately 200 ft. with a vertical rise of 65 ft. The lines from the ash chambers and the economizer ash hoppers are approximately 190 ft. long and have a vertical rise of 50 ft.

Ash Bunkers

Considerable choice of materials and design of ash bins and supports is available. A large number, perhaps most, are made of reinforced concrete. In some cases they are inside the boiler room and worked into the general design of coal bunkers, etc., such as illustrated in Fig. 14. They are occasionally built so as to span an alley, being supported by the buildings on each side.

Ash bunkers have also been built of brick, but unless very thick walls are used, they should either be buckstayed or have reinforcing bands laid up every five or six courses. The bricks should be hard and well burned, and laid in cement. The inner face of the walls should be of paving brick.

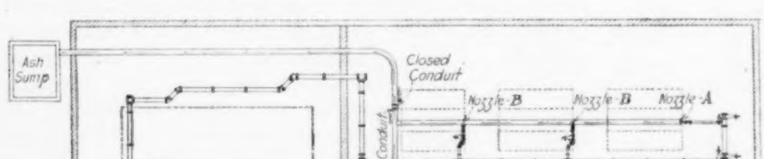

Fig. 21 Water Conveyor at Hell Gate Station. Plan

Hollow-tile ash tanks have several inherent advantages in that they are fireproof, prevent freezing of wet ash, are acidproof provide space for steel reinforcing rings within the wall, and have a smooth interior surface. The tiles are laid up in cement and are provided with key grooves on their joint faces.

Tanks may also be made up of a cast-iron skeleton filled in with cast-iron plates, the whole carried on a structural-steel framework. Cast iron, being much less subject to corrosion than steel, makes a durable tank for ash storage.

Water Conveyors

In systems of water conveyance the conduits carrying water are largely or wholly open. With a plentiful supply of water, this method has much to recommend it. There is no dust or heat and the ashes are carried away very quickly. Owing to the low velocity of the vehicle as compared with air, the wear is very small and usually is not troublesome. The conveyance, of course must always be down hill when open flumes are used, and a grade of 3 or even 4 per cent is desirable.

Where a natural supply of water from an elevator is not available a centrifugal pump may be used. The Hell Gate plant is an excellent instance of this kind, and the general

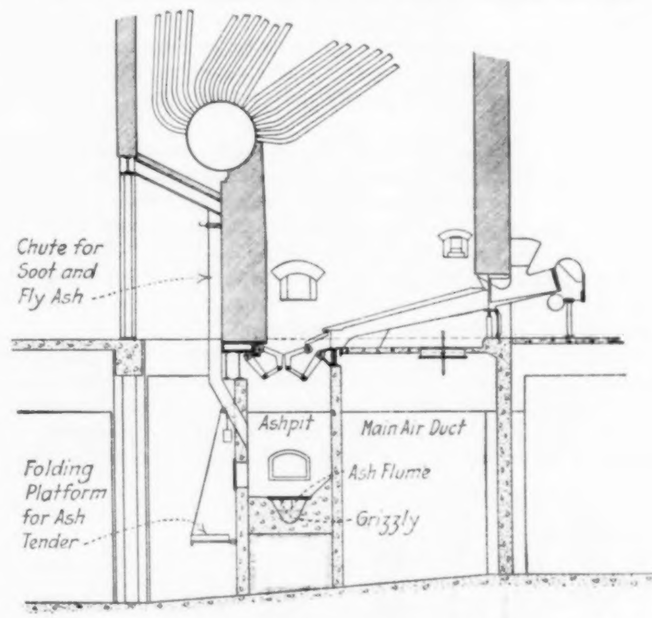

Fig. 22 Water Conveyor at Lacombe Station, Denver. Cross-Sectional Elevation

layout is shown in Fig. 21. This is a system of open flumes within the boiler room continuing through a closed pipe to the ash setting tank. Tributary flumes are carried along below each line of boilers and empty into a main cross-flume which runs along the boiler room wall near the turbine room. This main flume then turns and becomes a full pipe or enclosed conduit leading to a pit near the river side into which it discharges. The ashes are recovered from this pit by a grab bucket operated by a locomotive crane running on a track laid on the pier, and discharged into scows. The scows are towed out and dumped at sea about five miles east of Sandy Hook, or about thirty miles from the plant.

The flumes within the boiler room are of concrete with a bottom lining of vitrified earthen drain tiles or half-pipes. Those under the boilers are supported on structural-steel framework suspended from the firing floor, while the main cross-flume is carried on steel trestling.

A cross-section of the flume under the boilers is shown in Fig. 23. The ash as it leaves the clinker grinders drops directly into the water and is carried away. Access doors lined with common brick are provided so that any ob-

struction can be handled easily. These doors may be seen in Fig. 23.

The method of water supply is very interesting. There is a nozzle A at the head of each tributary flume, and an ingeniously arranged undercurrent nozzle B at the beginning of each succeeding ashpit. These undercurrent nozzles are at the bottom of the flume and are arranged to discharge horizontally downstream. They are placed to form steps in the flume so that the ash flows over them. At the head of the main conduit is a booster nozzle C.

The water for the various nozzles is taken from the circulating discharge tunnel and is supplied under pressure by 12-in. Lea-Courtenay centrifugal pumps direct-driven by 150-hp. Westinghouse motors. These pumps supply 5000 gal. per min. against a head of 75 ft., operating at 81.5 per cent efficiency.

The main pipe line which supplies the water nozzles has a bore of 16 in. and is arranged as a ring as clearly shown in Fig. 21.

Fig. 22 shows a cross-section of the flume under the boilers of the Lacombe station of the Denver Gas and Electric Light Company. The outstanding difference between this plant and the one at the Hell Gate station is that there are no clinker grinders. As a result, the flumes are protected by a "grizzly" composed of heavy bars set 6 in. center to center to withstand breaking up the large clinkers until they can drop between the bars into the flume. The water stream will easily handle clinkers about 12 in. square and 15 lb. in weight.

Water sprays cool the clinkers which are caught on the grizzly. Side-hinged doors 18 in. by 23 in. are provided through which the clinkers can be managed.

The station contains five boilers of 750 hp. each, operated at 250 to 300 per cent of rating. The coal is a sub-bituminous containing 6 to 7 per cent of ash. As the boilers are in line, the flume is single and straight. It is all of 2 per cent grade except the curve to the ash-settling tank, which is of 18-in. vitrified sewer pipe laid horizontally. The flume and sewer pipe is about 155 ft. total length. The capacity of the setting tank is 3000 cu. ft.

The water passes through a screen and is recirculated, and a 2-in. line is used to replenish occasionally the water in the system. The water is circulated by a 6-in. American open-runner centrifugal pump driven by a 20-hp. motor. The pump delivers about 1100 gal. per min. against a head of 25 ft. The system deals with about 33 tons of ashes in 24 hours.

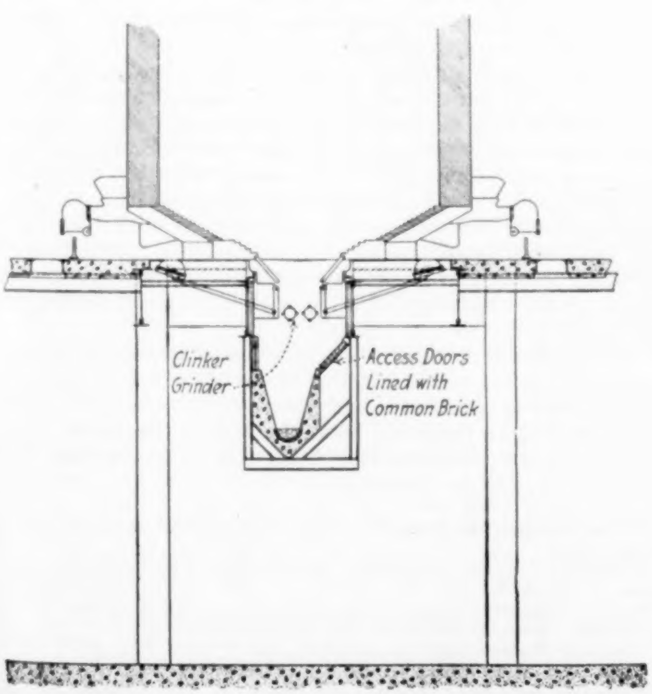

Fig. 23 Water Conveyor at Hell Gate Station. Cross-Sectional Elevation

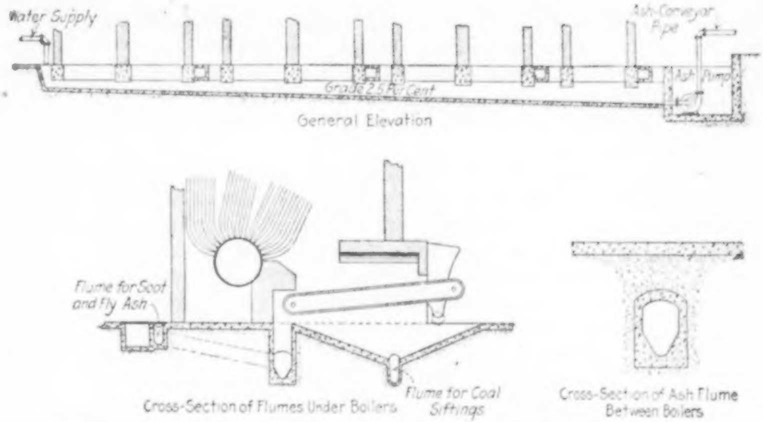

Fig. 24 Morrison's Water Conveyor at Great Western Sugar Refinery, Denver

An ingenious arrangement is that invented and patented by Mr. Morrison, of the Great Western Sugar Company of Denver, and installed in several of their plants. An example is partly illustrated in Fig. 24. A flume into which the ashes are fed is arranged under the boilers as in other water conveyors. This flume carries the ash-laden water into the suction connection of a centrifugal pump which discharges into the conveyor pipe. The conveyor pipe is of cast iron, 6 in. bore. Abrasion occurs slowly, but is confined to the bottom of the pipe. It is found that by rotating the conveyor pipes through an angle of 90 deg. every few years, maximum service is obtained out of each renewal.

In designing a water conveyor it must be borne in mind that while continuous dumping stokers such as chain grates or those equipped with clinker grinders may discharge dirctly into continuously running water, dumping stokers or any firing system where large clinkers are to be expected should discharge on to a grizzly of massive bars on which clinkers may be broken small. The water need only be running while dumping is in progress with stokers of the dumping type.

Concrete troughs, if carefully built, will be found to require very little repair. The inside lining should be smooth and free from pits. There should be a plentiful supply of water under a few pounds pressure. About 1000 gal. per min. should be supplied for each flume.

With natural draft, means should be devised to provide seals so as to prevent too great an excess of air from passing up through stoker dumps.

When the water is recirculated, the pump should be designed to handle gritty water, and occasional renewals due to this reason must be expected, though they will not be serious.

As there is often considerable dust and gas generated with stokers of the dumping type, it is advisable to provide the access doors with latches so that they cannot be blown open. Ample ventilation near where ashes are handled should always be provided.

In some localities in winter, trouble may be expected from water freezing in the bearings of grab buckets. In particularly cold situations the wet ash will freeze in the cars and cannot be dumped unless it is thawed out. Weather conditions may therefore prohibit water conveyors in some instances.

Discussion

In opening the extensive and extremely interesting discussion which followed the presentation of the paper, T. A. Marsh[1] said he would like to emphasize the serious effect of small ashpits and ash hoppers on stoker maintenance. Aside from the question of draft, he said, there was no one item which had caused more stoker failures than ashpit design. The cross-bar conveyor in water, shown in Fig. 16, had acquired great popularity in this country and he saw great possibilities for it. As the economic capacity of a man handling ash from an air conveyor was five to six tons, he saw no necessity of using larger than an 8-in. conveyor system. The remote-control steam valve for air conveyor systems, mentioned in the

[1] Chief Engineer, Green Engineering Co., East Chicago, Ind. Mem. Am.Soc.M.E.

paper, for shutting off the steam when the conveyor was not in use, was being received with great favor, and it appeared that the steam consumption of the average air conveyor might be cut in two by the use of this valve. It was very important, he said, to emphasize the author's caution in regard to keeping down the wear of steam nozzles in such systems.

F. B. Allen[1] discussed the design of ash hoppers which were bottom-dumped by means of power-controlled gates, discharging into some type of car beneath. It was essential for operating reasons, he said, to quench the ashes before dumping. The ordinary system of quenching by introducing one or two pipes perforated with holes was unsatisfactory as an excessive amount of water, with attendant difficulties, was essential for effective quenching. He then described, with the use of lantern slides, how better results could be obtained by running the quenching-water header outside of the hopper with a series of offtakes to quenching nozzles located through the hopper walls. Each quencher sprayed water in fine particles over the ash, and could be independently controlled and cleaned from the outside. He also gave particulars regarding the gates of this type of hopper.

R. H. Beaumont[2] said that in his opinion an ash-handling system should be capable of meeting the following conditions: (1) It must be capable of handling very large clinkers; (2) it must be of such design that abrasion has little or no effect on its parts, so that repairs are negligible; (3) it must not consume much power, nor waste it running idle; (4) it must be capable of lifting ashes to a good height without being subject to excessive wear by so doing; (5) it must handle either red-hot or dripping wet ashes with equal facility; (6) it must operate without making dust or exceptional noise; (7) it must have a high hourly capacity; (8) it must handle the general run of boiler-room refuse, such as soot, flue dust, broken bricks, grates, etc.; (9) it must accommodate itself to plant extension without complete remodeling; and (10) it must deliver ashes into a storage bunker so that they are acceptable as railroad freight. He described also the automatic electric skip hoist, which he considered the most important device used today in the handling of ashes.

Nevin E. Funk[3] said that ash-handling systems could be divided into two classes: those easy to operate and easily repaired without putting the entire apparatus out of commission; and those with no flexibility at all, which put at least one section of the boiler room out of commission in event of failure, and which cannot be aided by some crude makeshift in time of failure. In the first class he put the railroad car, the industrial railway with storage battery or trolley locomotives, and possibly the water-sealed ashpit of Fig. 10. In the second class he included all mechanical, air, and hydraulic conveyors. The use of railway cars did not mean raising the boiler room much more than required by many other schemes. He pointed out that even with a continuous ash-disposal system there must be sufficient storage to take care of periods of breakdown. He criticised a system which was used for both coal and ash because of the chance of getting ashes in the coal. He did not like the water-seal chain and cross-bar conveyor of Fig. 16 because of the liability to breakage. Even with two such systems parallel the breakage of one might easily result in the breakage of both; and it would not be easy to make repairs to such a system while the boilers were in operation. Speaking of the air-conveyor system with vacuum storage tank shown in Fig. 17, he said that there was danger of an explosion in such a system as gas might easily be generated in the tank. He was sure that hydraulic systems, even those depending upon flotation, would soon wear out due to abrasive action of the water and ashes.

Nixon W. Elmer[4] spoke of the problem of ash disposal in a powdered fuel plant.

I. E. Moultrop[5] said that of the three great handling problems in the power plant-handling water, coal, and ashes—that of handling ashes was the greatest. He spoke of the cost of disposing of the ashes after they had been removed from the power plant. A very large ashpit, he said, was absolutely essential. Ashes should be taken away with the use of as little machinery as possible, as one did not want machinery which, in case of breakdown, would interfere with the removal of ashes. He agreed with Mr. Funk that the best scheme was the industrial or steam railway because, in case of breakdown, the ashes could be dumped on the floor, later to be carried away.

[1] New York City.

[2] Pres., R. H. Beaumont Co., E. Chicago, Ill. Mem. Am.Soc.M.E.

[3] Operating Engineer, Philadelphia Elec. Co., Philadelphia, Pa. Mem. Am.Soc.M.E.

[4] Cons. Conveyor Engineer, Quincy, Mass. Mem. Am.Soc.M.E.

[5] Asst. Supt. Construction Bureau, Edison Elec. Illum. Co., Boston, Mass. Mem. Am.Soc.M.E.

T. Maynz[1] spoke of the necessity for ash storage space. In his plant an industrial railway with skip hoist to storage had been installed. In case of breakdown the ashes are dumped on the floor. The problem, he said, was not to cut down the head room of ashpits, but to give them all the room possible.

Sam H. Libby[2] spoke of some of the difficulties confronting the designer of hoists for ash-handling systems which resulted from incomplete or incorrect information about the details of what was expected of his designs.

E. H. Tenney[3] wrote in part: "In connection with the use of air ejector systems in downtown districts we found that, other things being equal, a great deal of consideration had to be given to the prevention of dust. In the particular installation in mind this was eliminated by the careful design of water spray rings and the proper installation of a standpipe vent on the ash-receiving tank. We have also found that where ash hoppers are located outside the plant so that trucks or other conveyances can drive beneath them, for loading purposes, considerable difficulty was encountered in the winter time, due to the ash gates freezing. This difficulty was easily overcome by the installation of small steam lines on all such ash-hopper gates for the purposes of thawing them loose. This point should not be lost sight of in building overhead ash hoppers where the question of hurried ash removal is necessary, as is the case with office-building or hotel power plants."

Rankin Eastin[4] sent in a description of the ash-sluicing system which he had installed at the Tell City Water & Light Company, Tell City, Ind.

Charles E. Prout[5] described the G. & G. telescopic hoist which is adaptable to ash-removal systems in hotels and office buildings when the ashes must be elevated to the street from a boiler room below ground level. This system was applicable, he said, to two conditions found with such types of building: those in which a wagon or railway car could be brought alongside and the cans emptied without removing them from the hoist, and those in which the cans could not be dumped without removal. The average can, he said, weighed about 150 lb. and could be elevated and returned to the basement at a speed of 60 ft. per min.

Eugene Hahn[6] presented an extensive discussion of the problem of removing ashes on shipboard.

George G. Bell[7] presented a written discussion in which he showed how the necessity of ash-handling equipment became acute when coal of high ash content and low fusing temperature was used for power purposes. He spoke also of the difficulty of proper quenching of the cinders and of the gases produced in the ashpit. He emphasized the need of a large ash-storage capacity to tide over difficulties with the ash-handling apparatus or with labor. He described, with lantern slides, the water-sealed type of ashpit originally proposed by Frederick Sargent for the Springdale plant and which was being installed in the extension of the Windsor plant of the West Penn Power Company.

In his closure to the discussion, John Hunter touched on some of the points brought up but said that he did not wish to enter into a detailed reply until he had an opportunity to do so in writing. He was sorry that his paper had been submitted to the printer before he had been able to include data on the G. & G. system which had been mentioned in the discussion.

[1] Test. Engineer, Cleveland Elec. Illum. Co., Cleveland, Ohio. Jun. Mem. Am.Soc.M.E.

[2] Managing Engineer, Hoist Dept., Sprague Elec. Wks., Bloomfield, N. J. Mem. Am.Soc.M.E.

[3] Chief Engineer of Power Plants, Union Elec. Light & Power Co., St. Louis, Mo. Mem. Am.Soc.M.E.

[4] Tell City Water & Light Co., Tell City, Ind.

[5] Adv. Mgr., Gillis & Geoghegan, New York, N. Y.

[6] Vice Pres. and Chief Engineer, Victor Engrg. Co., Inc., Philadelphia, Pa.

[7] West Penn. Power Co., Pittsburgh, Pa.

Stresses in Electric-Railway Motor Pinions

Determination of Their Distribution by the Photo-Elastic Method

By PAUL HEYMANS,[1] CAMBRIDGE, MASS., AND A. L. KIMBALL, JR.,[2] SCHENECTADY, N. Y.

This paper embodies some results of a general scientific study undertaken by the General Electric Company for the development of superior electric railway motor pinions. The work described was performed at the Massachusetts Institute of Technology, using the General Electric Companys apparatus for stress determination in transparent models by the photo-elastic method. Some of the supplementary mechanical tests were made at Schenectady, and throughout the work close contact was maintained with the Railway Motor Department and the Research Laboratory at Schenectady. A brief description and discussion of the photo-elastic method is first given, following which the stress distribution in, and the causes of ruptures of, given types of gear pinions used in electric-railway motors, as investigated by the photo-elastic method, are reported upon and discussed.

THE state of stress at any point in a solid body is determined when the traction across every plane through the point is known. There exist at any point three orthogonal planes across which the traction is purely normal and which are called the planes of principal stress. The normal tractions across those planes are called the principal stresses. The state of stress at any point is completely determined by the direction and the magnitude of the principal stresses at the point under consideration. The principal stresses, given in direction and in magnitude, express in the most general and complete way the elastic state at any given point. The bending moment, the shearing forces, etc., are readily deduced from the direction and the magnitude of the principal stresses. Furthermore, one of the principal stresses always expresses the maximum stress.

The notion of principal stress may be illustrated as follows:

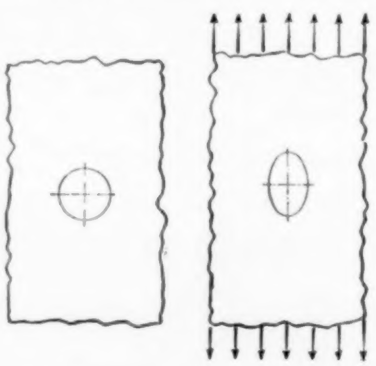

Fig. 1 Ellipsoidal Element Resulting from Subjecting a Spherical Element to Stress

Consider a spherical element in a solid body. External applied loads will deform this spherical element into an ellipsoidal element (Fig. 1). The axes of this ellipsoid will correspond in direction and in magnitude to the direction and the magnitude of the principal stresses.

The orientation and the form of the ellipsoid and therefore the direction and the magnitude of the principal stresses, will define the state of stress at the point under consideration.

The axes of the ellipsoid represent the largest and the smallest deformation at the point under examination. Correspondingly, the principal stresses give the direction and the magnitude of the maximum and the minimum stress.

If the three principal stresses vary from point to point in the structure, the problem to be dealt with is a three-dimensional elastic one. If one of the three principal stresses vanishes throughout, it is a two-dimensional elastic or plane-stress problem.

Corresponding to the three- and two-dimensional elastic-stress problems there are also the three- and two-dimensional elastic-strain problems, when the deformations corresponding to the principal stresses are considered.[1]

A great number of structural problems (bridge, ship, airplane, plate, dam, etc., construction) are, or their stress analysis may be reduced to, two-dimensional elastic problems.

The Photo-Elastic Method of Stress Determination

As set forth at the beginning of the paper, the state of stress at any point is most completely defined by the direction and the magnitude of the principal stresses. These are, therefore, the elements which we wish to determine for a complete analysis.

The photo-elastic method solves the two-dimensional elastic

Fig. 2 Representation of Colored Image When Both Normal Inside Pressure and Maximum Torque are Applied

(The darkest portions of the image reproduced above are purplish in the original; the next lighter portions, bluish green; the next lighter, reddish orange; and the lightest, yellow.)

problems. It primarily takes advantage of the double refracting properties shown by isotropic transparent substances when put under stress. The stresses in the structure may therefore be determined from models made of a homogeneous transparent material, and ordinarily on a reduced scale. The stresses in a steel, cement, or any other structure, homogeneous throughout and obeying Hooke's law of linear proportionality between stress and strain, may be readily deduced from the values obtained by the analysis of the corresponding transparent model for the case of two-dimensional elastic problems.

If plane polarized light is passed through a stressed specimen of celluloid and afterward through a second nicol prism whose principal section is parallel to the plane of polarization of the original beam of light, only the points where the principal stresses are respectively parallel and perpendicular to the principal sections of the crossed nicols remain dark. The result makes it possible to determine the directions of the principal stresses at any given point.

If now circularly polarized light be passed through the specimen, by interference of the two component rays, which in the double-refracting specimen have suffered a relative retardation at each point proportional to the difference in magnitude of the two principal stresses, a colored image is obtained. Practical considerations make it impossible to reproduce these colored images here, but an idea of their general appearance may be gained from Fig. 2.

By a comparison method, based upon the interposition in the proper direction of a comparison member of constant cross-section, put under uniform tension in a suitable frame (Fig. 3), the value of the difference of the principal stresses at any given point may be read on the dynamometer of the frame.

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[1] A complete theory of stress and strain may be found in the Treatise on the Mathematical Theory of Elasticity by A. E. H. Love, 3rd ed , chapters i–iv.

Now, in the two-dimensional elastic problems the transverse deformation, i.e., the deformation along a normal to the plane of the two principal stresses, is proportional to the sum of those two stresses.

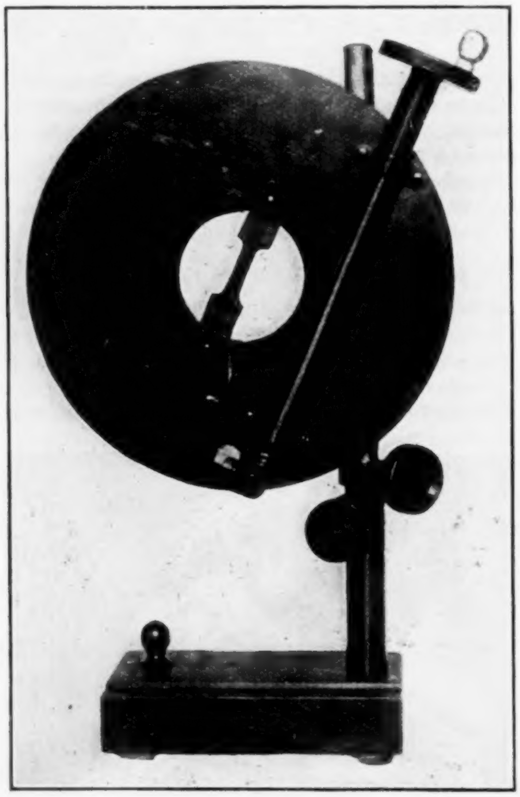

FIG. 3 FRAME FOR COMPARISON MEMBER DESIGNED BY E. G. COKER AND A. L. KIMBALL, JR.

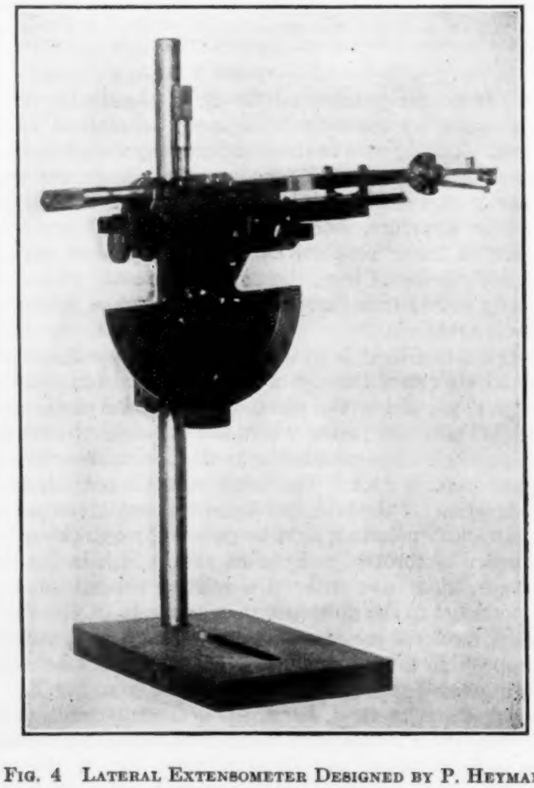

FIG. 4 LATERAL EXTENSOMETER DESIGNED BY P. HEYMANS

By means of a lateral extensiometer, Fig. 4, we measure this transverse deformation. From the values of the differences and the sums of the principal stresses, the separate values of each of them are computed, thus determining completely the state of stress.

A question naturally arising is whether the results obtained on a transparent body such as celluloid hold for structural materials. It is shown by the general discussion of the equations of elastic equilibrium that in the case of strain or plane stress in an isotropic body obeying Hooke's law of linear proportionality between stress and strain, the stress distribution is independent of the moduli of elasticity and consequently of the material of which the body is made. Thus the stress distribution experimentally determined in the case of a celluloid body is the same as it is when the body is made of any other isotropic substance such as iron, steel, etc., obeying Hooke's law, in distribution, direction, and magnitude. Moreover these conclusions derived from the general theory of elasticity have been checked by experiment.

The photo-elastic method can be applied to the great majority of structural problems, not only in taking the place of mathematical computation, but particularly in solving those structural problems where mathematics becomes too involved to be of help. Moreover it has the great advantage of giving the maximum stress at each point throughout the whole structure, and it therefore offers an effective means of increasing safety and reducing superfluous material.

STRESS DISTRIBUTION IN GEAR PINIONS

When accidents occur with gear wheels, besides the metallurgi-

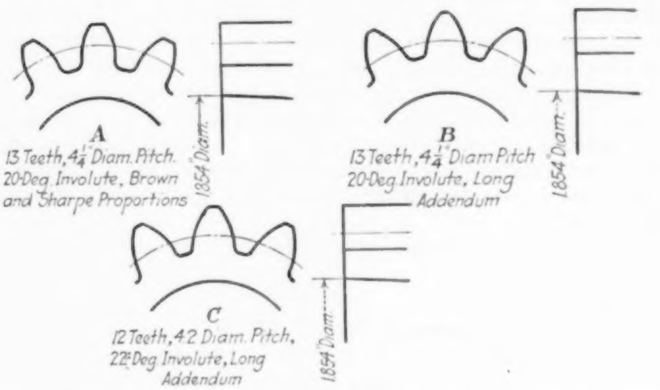

FIG. 5 TOOTH FORMS OF PINIONS SUBJECTED TO PHOTO-ELASTIC ANALYSIS

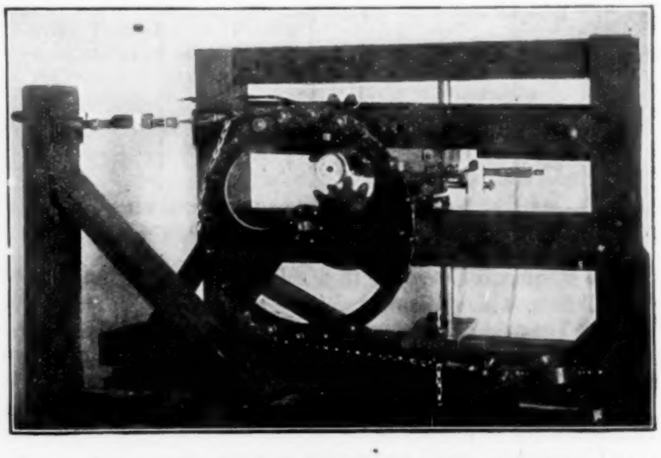

FIG. 6 FRAME USED FOR APPLYING LOADS TO CELLULOID MODELS OF PINIONS

cal question, three possible causes of failure suggest themselves:

a The gear wheel may not have been properly designed
b It may have failed under an excessive load
c When the pinion was shrunk hot or forced on to a tapered shaft, an excessive inside radial pressure may have been set up.

It is easy to see that the ordinary methods of resistance calculation of gear wheels, based on considering the tooth as a cantilever loaded at its end, would not be expected to give reliable and complete information as to stress distribution, not even for the root section of the tooth which is under consideration.

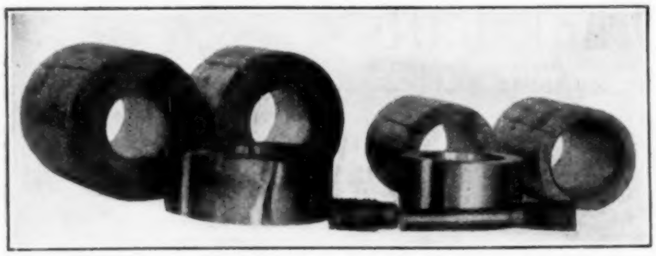

FIG. 7 STEEL RINGS RUPTURED BY BEING FORCED ON TO A TAPERED PLUG

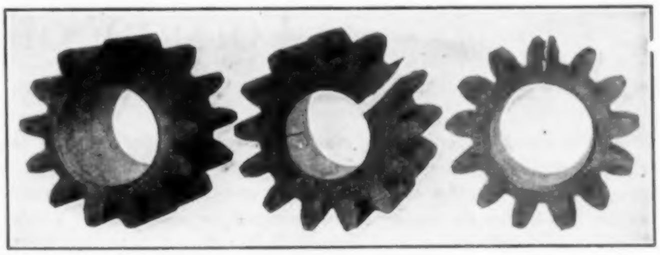

FIG. 8 STEEL PINIONS RUPTURED BY BEING FORCED ON TO A TAPERED PLUG

Indeed the shape of the tooth, the curvature at the root, the ratio of the diameter of the pinion bore to the root and outside diameter, the permanent stresses introduced by the placing of the pinion on the shaft, etc., all affect the stress distribution and the maximum stress. Photo-elastic analysis shows that these factors affect the stresses considerably more than would be expected from present methods of estimating. For standardized pinions the correction coefficients can only partially take account of these factors. For special pinions or for pinions of which more efficient running is required, a photo-elastic analysis seems to be the best if not the only effective way to determine the stress distribution and to locate the maximum stress.

Certain interesting points have been brought out by photo-elastic analysis which have been checked by tests carried out on steel sections. These are particularly interesting because they were unexpected.

Besides the stress distribution in the different sections of the pinions represented by Fig. 5 the photo-elastic analysis has given as maximum stress under normal inside radial pressure and normal torque:

80,000 lb. per sq. in. for tooth form *A*
70,350 lb. per sq. in. for tooth form *B*
60,900 lb. per sq. in. for the tooth *C*.

Moreover the 12-tooth pinion shows, besides a smaller maximum stress, a better stress distribution. Fig. 6 represents the frame used for the loading of the models. A tapered expansion ring is used to produce the radial inside pressure. The torque is measured by properly mounted dynamometers.

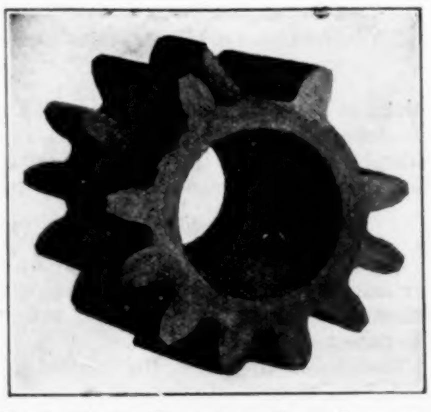

FIG. 9 FATIGUE FAILURES OF TEETH PRODUCED BY EXPERIMENT (WITH RADIAL PRESSURE IN BORE)

For steel pinions the maximum stress attained under normal conditions, although high, appears not to be excessive. *Tooth C appeared to be a better design* under normal conditions.

The stresses due to shrinking or forcing the pinion on the shaft can only be estimated. The pinion may be assumed to be a plain circular ring, for which case the stresses may be mathematically computed. The stress at any point of the ring as well as the maximum stress in the ring depends upon the lengths of the inside and outside radii. The opinion generally expressed is that for the case of the pinion the maximum stress will be intermediate between the maximum values obtained for rings of which the outside diameters are respectively equal to the root diameter of the tooth and to the outside diameter of the pinion, the inside bore being the same.

Photo-elastic analysis shows that *the gear pinion is even weaker than the plain circular ring whose outside diameter is equal to the root diameter of the tooth.* The change of external profile, due to the presence of the teeth, although requiring an addition of material, weakens the structure.

Figs. 7 and 8 show the steel specimens after having been tested by forcing a tapered plug into the bore; and Table 1 gives the rupture load applied to the tapered arbor forced into the bore for the different specimens. These confirm the photo-elastic results.

Previous to the photo-elastic investigation of the stresses due to radial inside pressure in pinion sections, fracture due to pure radial inside pressure would have been expected to occur through the minimum radial cross-section.

From the color image obtained in the photo-elastic analysis when normal inside pressure alone was applied, it appears that *the regions under the teeth are under higher stress* and that the points at the inside boundary right under the teeth are points of maximum stress. The regions in question in the color image mentioned are similar in appearance to the region *A* in Fig. 2.

TABLE 1 RUPTURE LOAD ON ARBOR FORCED INTO SPECIMENS TESTED

	Inside diam., in.	Outside diam., in.	Root diam., in.	Rupture load, lb.
Ring	1.854	3.5	...	85,000
Ring	1.854	2.5	...	51,000
Pinion	1.854	3.5	2.5	47,000

Fig. 8 gives the fractures obtained on steel sections. Two of the sections show fractures right through the thickest layer of material, while all of them started at points where the photo-elastic analysis had revealed maximum stress. The uneveness of the material must account for the deviation of the fracture in one of the cases.

Can any statement be made as to the causes of the failure by inspection of the shape of the fracture? In the case in which the authors were interested, the photo-elastic analysis determined the best design. As before said, either the placing of the pinion on the shaft, if carelessly done, for instance by pounding the pinion heavily on the tapered shaft, or excessive torque and blows due to sudden meshing or the taking on of a heavy load, will set up dangerous stresses.

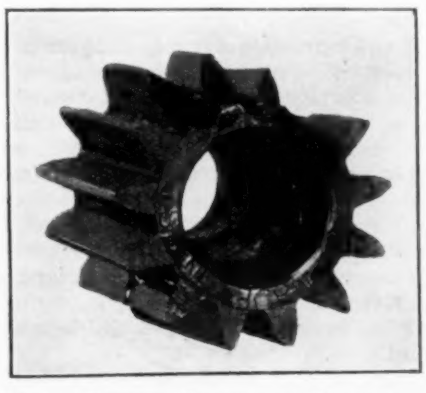

FIG. 10 FATIGUE FAILURES OF TEETH PRODUCED BY EXPERIMENT (WITH HEAVY RADIAL PRESSURE IN BORE). NOTE FRACTURE IS OF AN OPEN V-SHAPE

The authors' photo-elastic analysis has shown that *the sections of dangerous stresses are different for different values of inside radial pressure and applied torque load.*

The fracture shown in Fig. 9 is of an *open* V-shape. Photo-elastic analysis shows that *the higher the inside radial pressure becomes,* for a given torque load, *the sharper becomes the V-shape of the section of dangerous stresses.* (Fig. 10.) If the fracture is due to too high a torque load the angle of the V will approach 180 deg. Tests on steel sections have been made with a specially built impact machine.

Without inside radial pressure the fracture obtained is a straight line through the root section of the tooth. With increasing pressures the V-shaped fracture becomes sharper. For an inside radial pressure exceeding the elastic limit, however, the observation does not hold. The reason for this departure from what the photo-

(*Continued on page 137*)

Torsion of Crankshafts

By S. Timoshenko,[1] Philadelphia, Pa.

In order to apply the dynamics of elastic systems to the design of engines it is necessary to determine the torsional properties of the crankshaft. In the present paper the author considers the case of a crankshaft with a single throw and establishes the mathematical relations for such a case. He investigates three conditions of such a crankshaft: (1) no constraint, corresponding to ample clearance in the bearings; (2) complete constraint, corresponding to no clearance in the bearings; and (3) partial constraint, corresponding to ample clearance in the halves of the bearings nearest the web and no clearance in the other halves.

ENGINE design no longer comprises merely the application of statics, more attention being now paid to the fact that in various types of engines there is not even an approximation of a state of equilibrium. If there are variations in the torque and in the magnitudes of the force, they are necessarily cyclic, and as such they produce cyclic changes in the state of motion and deformation of the parts. The problem is no longer one of statics, but concerns the dynamics of an elastic system. Variations in the state of motion are associated with inertia forces, and the stresses in the various parts of the engine are no longer due to the actually impressed forces alone. Obviously, calculations which consider only the impressed forces must lead to erroneous results, and frequently a break occurs where the designer thought there was an ample margin of strength. Moreover cyclic changes in the state of deformation of a crankshaft, for example, bring about a cyclic change in the orientation of the various cranks, and the balance of the reciprocating parts, which may exist without such changes, is totally destroyed. With reciprocating engines the cyclic changes in the state of motion, referred to above, concern principally changes of the velocity of rotation. In order to apply the dynamics of elastic systems to the type of engine just mentioned, it is necessary to determine the torsional properties of the crankshaft. This is the object of the present paper, which deals with the simpler cases and will be followed by another discussing the two- and three-cylinder two-bearing crankshaft, the multi-throw crankshaft, and the effect of clearances in the bearings.

On account of the very complex structure of reciprocating engines, the calculation of the torsional vibrations in their crankshafts is impossible without making some simplifying assumptions: namely, that certain parts, such as shafts, are considered to be elastic, and others, such as flywheels, armatures of generators, etc., are considered to be absolutely rigid. With these assumptions the engine may be reduced to a system of flywheels situated on a shaft of uniform diameter. This shaft is called the "equivalent shaft;" its diameter may be chosen arbitrarily, but its length between each two flywheels must be such that it is equivalent as to torsion to the actual shaft between corresponding parts of the actual engine. That is to say, a given twisting moment M must produce the same torsional deformation in both.

The length of the equivalent shaft is called the reduced length l_0. If we denote by the torsional rigidity C of a shaft the product of the modulus of shear G into the quantity which measures its resistance to twist, then by elementary mechanics we have for the twist δ of a straight shaft of length l of whatsoever cross-section (provided it is uniform),

$$\delta = \frac{Ml}{C} \qquad [1]$$

In the above, as well as in the following, angles will be expressed in radians and all other units in inches and pounds.

For a circular cross-section,

$$C = G\theta \qquad [2]$$

wherein $\theta = \pi d^4/32$ is the polar moment of inertia of the cross-section.

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Contributed by the Research Committee and presented at the Annual Meeting, New York, N. Y., December 4 to 7, 1922, of The American Society of Mechanical Engineers. Abridged. All papers are subject to revision.

Similarly, we have for the equivalent shaft:

$$\delta = \frac{Ml_0}{C_0} \text{ and } C_0 = G\theta_0 \qquad [3]$$

from which—

$$l_0 = \frac{C_0 l}{C} = \frac{\theta_0 l}{\theta} \qquad [4]$$

As the foregoing shows, the calculation of the reduced lengths of the equivalent shaft has no difficulty for those portions of shaft in the actual engine which are straight. When the actual shaft is a crankshaft, however, the situation becomes very much more involved since it becomes necessary to determine the angular deformation δ brought about by a twisting moment M. Owing to the complex geometric shape of a crankshaft, these calculations can be accomplished approximately only. Various assumptions will have to be made, of which some are only roughly true. It is assumed that the throw is built up of component shapes whose deformations are totally independent of each other. Further, it is necessary to make certain assumptions about the nature and amount of con-

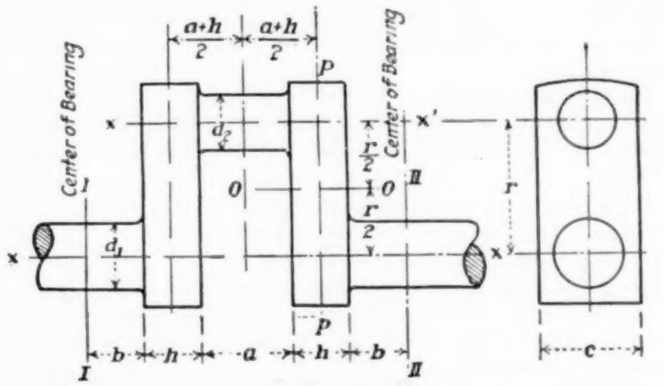


Fig. 1 Conventional Type of Single-Cylinder, Two-Bearing Crankshaft

straint in the bearings. In the present paper only the single-throw crankshaft (Fig. 1) will be considered, and therefore the very considerable influence on it of the neighboring cranks will be neglected. Calculations will be made with three degrees of constraint.

Definitions and Symbols

Fig. 1 shows the conventional type of a single-cylinder, two-bearing crankshaft. In addition to the dimensions shown thereon, the following is a list of the definitions and principal symbols used in the paper:

Torsional rigidity, generally denoted by C, is the product of the modulus of shear G into the quantity corresponding to its resistance to twist. For a circular cross-section the latter is the polar moment of inertia of the cross-section in respect to the axis of twist. For a rectangular cross-section the quantity referred to is more complex. The formula usually used is only approximately true.

Flexural rigidity is similarly the product of Young's modulus into the quantity I corresponding to the resistance of the cross-section against bending. The latter is always the equatorial moment of inertia of the cross-section with respect to the axis of bending.

$C_0 = G\theta_0 = \dfrac{\pi}{32} d_0^4 G$ = torsional rigidity of equivalent shaft

$C_1 = G\theta_1 = \dfrac{\pi}{32} d_1^4 G$ = torsional rigidity of journal

$C_2 = G\theta_2 = \dfrac{\pi}{32} d_2^4 G$ = torsional rigidity of crankpin

$C_3 = G\theta_3$ = torsional rigidity of web in respect to twist around

O-O. Owing to the junction of journal and pin to the web, the latter's cross-section is not clearly defined, but in order to allow for local stresses at these junctions, at least in some measure, the assumption is made that the cross-section is a rectangle with sides r and c, whence—

$$\theta_3 = \frac{c^3r^3}{3.6(c^2 + r^2)}$$

$C_3' = G\theta_3'$ = torsional rigidity of web with respect to twist around p-p. The cross-section is rectangular with sides h and c, whence—

$$\theta_3' = \frac{c^3h^3}{3.6(c^2 + h^2)}$$

$B_1 = EI_1 = \frac{\pi}{64} d_1^4E$ = flexural rigidity of journal

$B_2 = EI_2 = \frac{\pi}{64} d_2^4E$ = flexural rigidity of crankpin

$B_3 = EI_3 = \frac{hc^3}{12} E$ = flexural rigidity of web against bending in the plane through p-p perpendicular to the plane of the drawing of Fig. 1.

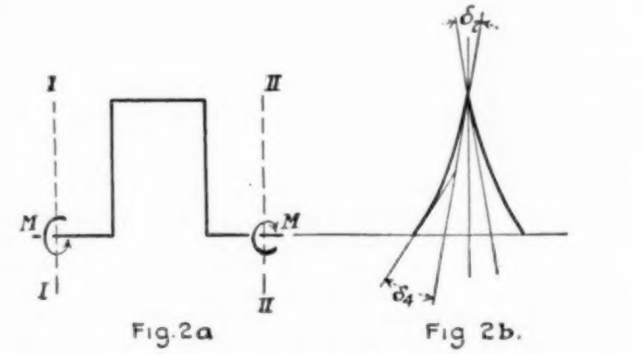


FIG. 2a DIAGRAMMATIC REPRESENTATION OF SINGLE-THROW CRANKSHAFT BEFORE DEFORMATION

FIG. 2b DIAGRAMMATIC REPRESENTATION OF SIDE VIEW OF DEFORMED CRANKSHAFT

$F_1 = \frac{\pi d_1^2}{4}$ = area of cross-section of journal

$F_2 = \frac{\pi d_2^2}{4}$ = area of cross-section of pin

$F_3' = h \times c$ = area of cross-section of web taken on O-O

$F_3 = r \times c$ is taken to be the cross-section of the web on line p-p. This quantity is used in the calculation for the deformation due to shear in the plane through p-p perpendicular to the plane of Fig. 1. Similarly, as in the case of torsion, around O-O the cross-section is not clearly defined and $r \times c$ is used to allow for stresses at the junction of pin and journal to the web.

CASE I—NO CONSTRAINT, CORRESPONDING TO AMPLE CLEARANCE IN THE BEARINGS

Fig. 2a is a diagrammatic representation of the throw before deformation with a twisting moment M applied at the middle cross-section of the journals. Fig. 2b is a side view of the deformed crankshaft. The total twist consists of the sum of the deformations of the portions b (see Fig. 1) of the journals, of the crankpin and of the two webs.

In the complete paper it is shown that—

$$l_0 = \frac{2bC_0}{C_1} + \frac{aC_0}{C_2} + \frac{2hC_0}{C_3} + \frac{2rC_0}{B_3} \qquad [5]$$

CASE II—CONSTRAINT AT BEARINGS COMPLETE, CORRESPONDING TO NO CLEARANCE

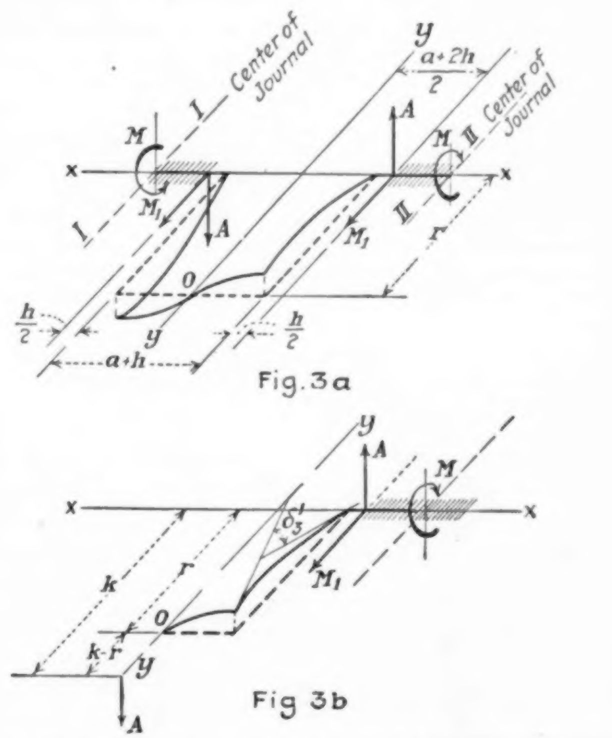
The deformed crank is shown diagrammatically in Figs. 3a and 3b. The constraint gives rise to a force A and a moment represented by the vector M_1 at each of the journals. The forces A and moments M_1 act in a plane through the axis of the journals perpendicular to the plane of the throw.

It is shown in the complete paper that—

$$l_0 = \left\{\frac{2bC_0}{C_1} + a\left(1 - \frac{r}{k}\right)\frac{C_0}{C_2} + 2h\left(1 - \frac{r}{2k}\right)\frac{C_0}{C_3} + 2r\left(1 - \frac{r}{2k}\right)\frac{C_0}{B_3}\right\} \qquad [6]$$

from which it is seen that the reduced length of the throw is completely determined if we know the fraction r/k. For $A = 0$ or $k = \infty$ we must have the case of no constraint and, indeed, by placing $k = \infty$ in [6] we obtain [5]. The effect of the constraint is therefore measured by the quantity k. And in the special case where $2b = a$ and $C_1 = C_2$ we can immediately compare the case of a complete constraint with no restraint and find that the constraint decreases the reduced length in the ratio of 1 to $[1 - (r/2k)]$.

The author then proceeds to determine Equation [7], which makes it possible to calculate k, whence l_0 is obtained from [6]. γ is a coefficient having the value 1.2.



FIGS. 3a AND 3b DIAGRAMMATIC REPRESENTATION OF DEFORMED CRANK—CASE II

$$k = \frac{\frac{r(a+h)^2}{4C_3'} + \frac{ar^2}{2C_2} + \frac{a^3}{24B_2} + \frac{r^3}{3B_3} + \frac{hr^2}{4C_3} + \frac{\gamma a}{2F_2G} + \frac{\gamma r}{F_3'G} + \frac{\gamma h}{F_3G}}{\frac{ar}{2C_2} + \frac{r^2}{2B_3} + \frac{hr}{2C_3}} \qquad [7]$$

This equation is at variance with the equation given by Geiger (Ueber Verdrehungsschwingungen von Wellen), for which reason, and as a check on the above results, the derivation of k and l by means of Castigliano's theorem is given in an appendix to the complete paper. This method is particularly useful in more complicated cases where the deformation of the throw is not as easily seen as in the preceding case.

CASE III—PARTIAL CONSTRAINT

It is here supposed that there is ample clearance in the halves of the bearings nearest the web, and no clearance in the other half. See Figs. 4a, 4b and 4c. In this manner we will now have bending, as well as torsion, in the halves of the journal next to the webs, a condition which no doubt prevails in practice. The author finds that—

$$k = \frac{\dfrac{r(a+h)^2}{4C_3'} + \dfrac{ar^2}{2C_2} + \dfrac{a^3}{24B_2} + \dfrac{r^3}{3B_3} + \dfrac{hr^2}{4C_3} + \dfrac{\gamma a}{2F_2G} + \dfrac{\gamma r}{F_3'G} + \dfrac{\gamma h}{F_3G}}{\dfrac{ar}{2C_2} + \dfrac{r^2}{2B_3} + \dfrac{hr}{2C_3}}$$

$$+ \frac{b^3}{3B_1} + \frac{b^2(a+2h)}{2B_1} + \frac{b(a+2h)^2}{4B_1} + \frac{\gamma b}{F_1G} \dots \quad [8]$$

Numerical Example

Let $d_1 = d_2 = 10.25$ in.; $a = 13$ in.; $r = 11$ in.; $b = a/2 = 6.5$ in.; $h = 5.5$ in.; $c = 14$ in. Then—

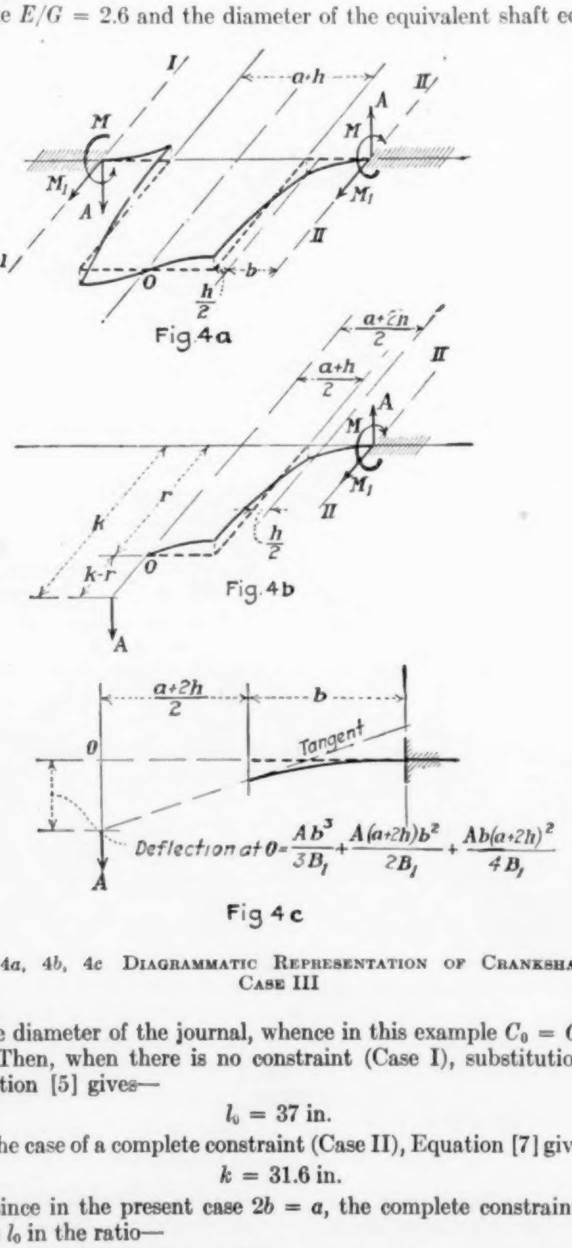
$$B_1 = B_2 = \frac{\pi \times 10.25^4 E}{64} = 54E; \quad B_3 = \frac{5.5 \times 14^3 E}{12} = 1258E$$

$$C_1 = C_2 = \frac{\pi \times 10.25^4}{32} G = 1085G;$$

$$C_3' = \frac{14^3 \times 5.5^3}{3.6(14^2 + 5.5^2)} G = 562G$$

$$C_3 = \frac{14^3 \times 11^3}{3.6(14^2 + 11^2)} G = 3200\,G$$

Take $E/G = 2.6$ and the diameter of the equivalent shaft equal



Figs. 4a, 4b, 4c Diagrammatic Representation of Crankshaft—Case III

to the diameter of the journal, whence in this example $C_0 = C_1 = C_2$. Then, when there is no constraint (Case I), substitution in Equation [5] gives—

$$l_0 = 37 \text{ in.}$$

For the case of a complete constraint (Case II), Equation [7] gives—

$$k = 31.6 \text{ in.}$$

and since in the present case $2b = a$, the complete constraint reduces l_0 in the ratio—

$$1 : \left(1 - \frac{r}{2k}\right) = 1 : (1 - 0.174)$$

that is, the reduced length is reduced by 17.4 per cent.

In the case of a partial constraint (Case III), Equation [8] gives—

$$k = 44.2 \text{ in.}$$

and the reduced length, as compared with the case of no constraint, is reduced in the ratio of—

$$1 : \left(1 - \frac{11}{2 \times 44.2}\right)$$

that is, by 12.5 per cent.

These examples plainly show the effect of the constraint at the bearings. The more complete it is, the stiffer the shaft and the shorter the reduced length. The calculations further show that an increase in the diameters of journal and pin and an increase in the length of the pin cause an increase of k and thereby a reduction of the effect of the constraint and reduced bearing pressures A. On the other hand, an increase in the thickness of the web brings about a smaller length k and a corresponding augmentation of the bearing reactions.

Corrosion Problems

A PAPER on Corrosion as Affecting Metals Used in the Mechanical Arts was read on December 5, 1922, before the Sheffield Association of Metallurgists and Metallurgical Chemists, by Dr. W. H. Hatfield, Brown-Firth Research Laboratories, and past-president of the Association. The object of the paper was to give the particulars and results of an interesting set of experiments made at these laboratories to decide upon the resistance of typical industrial metals to various corroding agents. Dr. Hatfield referred to former work in the matter of various steel alloys, including that by Brearley in regard to stainless steel, and added that acid tests—the means generally employed—gave no direct indication, unfortunately, of the usefulness of an alloy for general corrosion-resisting purposes. A 30 per cent nickel steel was very resistant to sulphuric and hydrochloric acids, but was rapidly soluble in nitric acid of 1.20 specific gravity. The 14 per cent chromium steel was readily soluble in hydrochloric and sulphuric acids, but insoluble in nitric acid, while the 15 per cent silicon steel, although practically insoluble in all three acids, could not be considered completely rustless. These outstanding facts led largely to the carrying out of the experiments.

The materials dealt with numbered over twenty, and in addition to pure irons, carbon steels, alloy steels, cast-iron, they included nichrome and a good selection of non-ferrous metals.

He drew the two following conclusions: (1) There were now available steels and alloys which could effectively resist ordinary corroding influences. The different alloys, while at times each resisting the same media, did not give the same response to other media. It was essential in the application of corrosion- and acid-resisting alloys for actual practical experiments to be made in connection with the particular application intended, since not only composition, but also concentration, temperature, and extraneous influences had to be considered. (2) There was no obvious law, or set of laws, at present available, nor was any existing working theory sufficiently satisfactory to render effective aid to the investigator.

In regard to determining the relation which might exist between the electrode potential and solubility, it being understood that solubility was a function of electrode potential, the author said the main remarks to be put forward were: (1) When rapid solution took place the potential was highly negative; (2) when the potential had a high positive value very little corrosion or chemical action took place; and (3) otherwise the magnitude of the potential gave no indication of the solution properties of the metal. Another point worthy of note was the fact that in those cases in which rapid corrosion took place with evolution of gas, the potentials in hydrochloric and sulphuric acids were approximately the same, while in the case of the normal acids (acids containing 1 gram of available hydrogen per liter) the potential in nitric acid was about 0.2 volt less negative than in the other two acids. No satisfactory explanation of these observations could be put forward. The history of the metal would have some influence on the measured electrode potential, as shown by a table included in the paper. *Engineering*, Dec. 15, 1922, p. 747.

The Design of Cooling Towers

By C. S. Robinson,[1] Cambridge, Mass.

The author points out that engineers have designed cooling towers in the past on empirical information and in accordance with the experience of previous successful designs. Wide departure from standard designs is difficult because of the lack of scientific basis for the design. The author therefore establishes the general principle applicable to cooling-tower design and derives equations for the use of the designer. He presents a quantity of experimental data to substantiate the validity of his formulas and shows by an actual experiment how these formulas are applicable to the design of a counter-current cooling tower.

THE factors influencing the design of cooling towers have been studied by a number of investigators. There has been, however, no statement of the influence of these factors of such a nature that it has been possible to calculate easily the performance of a given tower under widely varying internal and external conditions. Engineers have found that towers of specified construction, when operated at specified air and water rates, may be expected to cool water to within a certain number of degrees of entering air tempeature and to discharge the air from the tower nearly saturated at some temperature approaching that of the entering water. In the absence of anything but empirical information, towers are therefore constructed along certain well-established and successful lines. However, this empirical knowledge will not enable the engineeer to predict what will happen when conditions depart widely from standard practice.

Investigations carried on by the Department of Chemical Engineering at the Massachusetts Institute of Technology under the direction of W. K. Lewis in connection with humidification and air drying, have led to the development of fundamental conceptions as to the mechanism involved in the transfer of heat between liquids and gases and in the vaporization and condensation of liquids and vapors. It is the purpose of this paper, which is based on the principles demonstrated in one written by Professor Lewis,[2] to show how these concepts can be applied to the particular case of cooling towers, and to devise by these means methods by which the engineer can simplify their design.

General Principles

There are two principles upon which all of the subsequent work will be based, (*a*) the conservation of matter and energy, and (*b*) the potential concept. The latter may be expressed briefly as the effect upon the rate of flow of matter or energy of the driving force applied.

The conservation of energy as applied to a cooling tower may best be shown by means of a heat balance. This can be written as—

$$s_w w(t_1 - t_0) = Ws'(T_1 - T_0) + Wr'(H_1 - H_0) \quad [1]$$

where s_w = average specific heat of the water between the bottom and top of the tower
- w = weight of water leaving tower
- s' = humid heat[3] of the air entering the tower, that is, the heat required to raise one degree in temperature one pound of dry air plus the water vapor H that it contains
- T = temperature of air
- t = temperature of water
- W = weight of air (moisture free) entering tower
- r' = total heat of water vapor at the top of the tower at the temperature of the leaving air minus the heat of the liquid of water at the temperature of the entering water.

The subscripts 1 and 0 refer to the top and bottom of the tower, respectively.

Equation [1] is used universally by engineers at the present time in cooling-tower problems, usually in the approximate form—

$$w(t_1 - t_0) = Ws(T_1 - T_0) + Wr(H_1 - H_0) \quad [2]$$

where s and r are the average humid heat and latent heat between the top and bottom of the tower, respectively.

The potential concept may be applied to both the rate of transfer of heat from the liquid to the gas and to the rate of diffusion of water vapor from the liquid to the gas, as shown in the following paragraphs.

The rate or flow of heat is proportional to the temperature difference between the liquid and the gas, and the rate of diffusion of water vapor is proportional to the difference between the vapor pressure of the liquid water and the partial pressure of the water vapor in the gas. The amount of heat flowing from the water to the gas per unit time would therefore be—

$$WsdT = haAdx(T - t) \quad [3]$$

where h is the coefficient of heat transfer per unit area (sq. ft.), a the square feet of cooling surface per cubic foot of volume of the tower, A the cross-sectional area of the tower, and x the height of the tower.

In the same way the weight of the vapor vaporizing per unit time would be—

$$WdH = k'aAdx(P' - p) \quad [4]$$

where k' is the diffusion coefficient in pounds per unit area of exposed surface, P' the vapor pressure of the water, and p the partial pressure of the water vapor in the air. Since for small partial pressures p is nearly proportional to the absolute humidity H, Equation [4] may be written as—

$$WdH = kaAdx(P - H) \quad [5]$$

where P, the vapor pressure of the water, is expressed in terms of absolute humidity, that is, the absolute humidity of saturated air at the temperature of the water in question.

Dividing [5] by [3] gives—

$$\frac{dH}{sdT} = \frac{k(P - H)}{h(T - t)} \quad [6]$$

The mechanism by which the heat passes from water to the air may be understood by considering that the heat flows first from the interior of the water to the surface, and then from the surface through a substantially stationary air film in contact with the surface to the moving air.

It has been shown[2] that $h/k = s$ when h refers to the coefficient of heat transfer through the air film only. If the total coefficient of heat transfer from the interior of the water to the moving air be used instead of h, the ratio h/k would not equal s, but would only approximate it to a greater or lesser degree according to whether the heat flow through the water took place easily or with difficulty as compared with the flow through the air film.

Unless otherwise noted h and k will henceforth refer to the overall coefficient of heat transfer and vapor diffusion, respectively.

Therefore, taking $h/k = s$, [6] becomes—

$$\frac{dH}{dT} = \frac{P - H}{T - t} \quad [7]$$

Equation [7], stated in words, says that the *differential* increase in humidity of the air is to the *differential* increase in temperature of the air as the humidity difference between the air and the water is to the temperature difference between the air and the water.

Equation [7] when integrated between proper limits would give the change in humidity and temperature of the water in its passage through the cooling tower. However, while the humidity of the air and the vapor pressure of the water are related to each other and the temperatures of the air and water are also related, the relationships are nevertheless of such a nature that exact integration is impossible, and an approximation is obtained by the following assumption.

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[1] Department of Chemical Engineering, Massachusetts Institute of Technology.

[2] W. K. Lewis, The Evaporation of a Liquid into a Gas, Mechanical Engineering, July, 1922, p. 445.

[3] Wm. S. Grosvenor, Trans. Am. Inst. Chem. Engrs., 1908.

Presented at the Annual Meeting, Dec. 4 to 7, 1922, of The American Society of Mechanical Engineers. Slightly abridged. All papers are subject to revision.

It will be noted that the temperature difference between the water and the air in a counter-current cooling tower does not change greatly between the top and bottom of the tower, nor does the difference in humidities between the water and air. It is therefore a justifiable assumption that the average temperature difference is approximately equal to the arithmetical mean temperature difference, and that the average humidity difference is approximately equal to the arithmetical mean humidity difference. It may therefore be said that for the whole tower the total increase in humidity of the air is to the total increase in temperature of the air as the arithmetical mean humidity difference between the air and the water is to the arithmetical mean temperature difference between the air and the water.

For the tower shown in Fig. 1 where the subscripts 0 and 1 represent conditions at the bottom and top, respectively, there may now be written—

$$\frac{H_1 - H_0}{T_1 - T_0} = \frac{H_1 - P_1 + H_0 - P_0}{T_1 - t_1 + T_0 - t_0} \qquad [8]$$

which is the integrated form of Equation [7].

Experimental Data

The author has collected and arranged in Table 1 the results of twenty-three tests made by himself and others on various types of cooling towers.

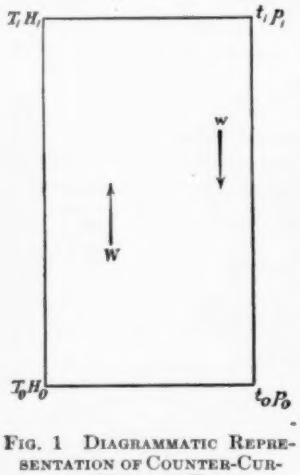

Fig. 1 Diagrammatic Representation of Counter-Current Cooling Tower

Tests 1 to 5 were on a slat type of forced-draft tower, 6 and 7 were on the same tower operating under natural draft, and Test 23 was on a Wheeler-Balke tower, all described in the Journal of the Ohio Society of Mechanical, Electrical and Steam Engineers, vol. 7. Tests 8 and 9 were on a forced-draft slat-type tower described in the Journal of the American Society of Refrigerating Engineers, vol. 3, 1916–17, p. 32. Test 10 was on a natural-draft tower described in the Transactions of The American Society of Mechanical Engineers, vol. 31, 1909, p. 75. Tests 11 to 22 were on an experimental Badger spray-type tower erected at the Massachusetts Institute of Technology, and the results have not been previously published.

The method of calculating the data may be explained by reference to Test 1.

T_0, (T_1) = temperature of inlet (outlet) air, deg. fahr.
t_1, (t_0) = temperature of inlet (outlet) water, deg. fahr.
Rel. Hum. = relative humidity in per cent
Q = heat given up by water in B.t.u. per min.
$= 651 \times 8.3 \times (105 - 84.7) = 110{,}500$
V = cubic feet of air per minute
H_0 = absolute humidity of inlet air in pounds of water per pound of dry air, as read from humidity chart[1]
P_0 = equivalent absolute humidity of outlet water, i.e., the absolute humidity of air saturated at the water temperature
H_1 = absolute humidity of outlet air, assuming saturation
P_1 = equivalent absolute humidity of inlet water
H = mean humidity difference between the air and water
$$= \frac{0.026 - 0.006 + 0.050 - 0.031}{2} = 0.0195$$
W = pounds of dry air per minute, as calculated from the volume of air V divided by the humid volume (the cubic feet occupied by one pound of dry air plus the water vapor which it contains, as read from the humidity chart)
w = pounds of water evaporated per minute
$= W(H_1 - H_0) = 3700(0.031 - 0.006) = 93$
$ka \times$ vol. = pounds of water evaporated in the whole tower per minute per one pound mean humidity difference
$$= \frac{93}{0.0195} = 4800$$
Q_{sens} = sensible heat picked up by the air per minute
$= Ws(T_1 - T_0)$, where s is the humid heat as read from the humidity chart[3] ($s = 0.25$)
$= 3700 \times 0.25(90 - 71) = 18{,}000$ B.t.u.
t = mean temperature difference between air and water
$= (105 - 90 + 84.7 - 71)/2 = 14.4$
$ha \times$ vol. = B.t.u. picked up as sensible heat by air in the whole tower per degree temperature difference
$$= \frac{18{,}000}{14.4} = 1250$$
$$\frac{h}{k} = \frac{ha \times \text{vol.}}{ka \times \text{vol.}} = \frac{1250}{4800} = 0.26$$
$ha = \dfrac{ha \times \text{vol.}}{\text{vol.}}$ where "vol." is the volume of the tower.

The dimensions of the tower in Test 1 were not given in the article, but other information led to the assumption that the tower was 10 × 10 × 22 ft., a volume of 2200 cu. ft., whence—

$$ha = \frac{1250}{2200} = 0.57 \text{ and } ka = \frac{4800}{2200} = 2.18$$

[1] Wm. S. Grosvenor, Trans. Am. Inst. Chem. Engrs., 1908.

TABLE 1 COOLING-TOWER TEST DATA

Test No.	Tower No.	T	T_1	t_1	t_0	Rel. hum.	Gal. per min.	Q	V
1	1	71	90	105	84.7	40	651	110,500	53,900
2	1	72	93	107.8	87.5	60	638	108,000	50,100
3	1	64	96	112	88.5	60	638	124,500	51,400
4	1	69	92	108.5	87	48	643	115,000	52,200
5	1	83	95	109.9	90.5	48	640	103,400	50,600
6	1	43	101	116	98	75	632	94,800	23,500
7	1	60	118	135	115.8	73	630	102,000	15,575
8	2	82	99	115	83	65	617[c]	176,800	72,700
9	2	75	96	109	81	74	668	168,800	72,700
10	3	25	80[a]	87.5	71.3	72	590	78,000	......
11	4	75.8	87.7	95.1	89.1	85	248	12,400	11,100[c]
12	4	77.0	103.9	112.1	99.2	78	230	24,700	9,060[c]
13	4	79.0	112.1	123.3	105.5	76	219	32,400	8,490[c]
14	4	76.1	89.9	94.4	87.9	60	288	15,500	9,700[c]
15	4	76.9	101.7	106.5	95.5	60	278	25,400	8,930[c]
16	4	77.0	112.9	123.7	105.2	57	259	39,800	8,970[c]
17	4	72.1	91.2	94.5	88.0	78	254	13,700	8,090[c]
18	4	73.0	105.1	113.7	99.9	75	246	28,100	9,030[c]
19	4	76.6	115.0	132.1	109.4	68	237	44,700	9,590[c]
20	4	72.9	90.9	92.4	88.4	87	269	8,900	6,160[c]
21	4	72.3	105.4	113.3	100.6	89	257	27,000	8,870[c]
22	4	72.7	115.2	128.2	108.1	88	241	40,200	8,800[c]
23	5	91	106	109	97	59	3,200	319,000	125,000[c]

Test No.	Tower No.	H_0	P_0	H_1	P_1	H	W	Δw	$ka \times$ vol.
1	1	0.006	0.026	0.031	0.050	0.0195	3700	93	4,800
2	1	0.010	0.028	0.034	0.055	0.0195	3410	82	4,200
3	1	0.008	0.029	0.037	0.062	0.023	3460	100	4,300
4	1	0.007	0.028	0.033	0.056	0.022	3560	89	4,000
5	1	0.012	0.031	0.036	0.058	0.0205	3420	82	4,000
6	1	0.004	0.040	0.044	0.071	0.0315	1550	62	2,000
7	1	0.008	0.070	0.075	0.129	0.057	915	61	1,100
8	2	0.015	0.024	0.041	0.068	0.018	5200[c]	135	7,500
9	2	0.014	0.023	0.037	0.057	0.0145	5276[c]	121	8,300
10	3	0.002	0.016	0.022	0.028	0.010	5200[c]	99.5	.....
11	4	0.015	0.030	0.027	0.036	0.012	790[c]	9.5	790
12	4	0.015	0.041	0.045	0.063	0.022	648[c]	19.4	880
13	4	0.016	0.051	0.059	0.090	0.033	606[c]	26.7	810
14	4	0.011	0.029	0.029	0.035	0.012	693[c]	12.5	1,040
15	4	0.011	0.037	0.043	0.052	0.018	638[c]	20.4	1,130
16	4	0.011	0.050	0.061	0.090	0.034	641[c]	32.0	940
17	4	0.013	0.029	0.031	0.036	0.0105	578[c]	10.4	990
18	4	0.013	0.042	0.047	0.066	0.024	644[c]	21.9	910
19	4	0.013	0.058	0.065	0.118	0.049	692[c]	36.0	730
20	4	0.015	0.029	0.030	0.033	0.0085	440[c]	6.6	780
21	4	0.015	0.043	0.048	0.065	0.023	623[c]	20.9	910
22	4	0.015	0.055	0.066	0.104	0.039	628[c]	32.0	820
23	5	0.019	0.039	0.051	0.057	0.013	8250[c]	273.0	21,000

Test No.	Tower No.	Q_{sens}	Δt	$ha \times$ vol.	h/k	ha	ka	u	$100 \times \frac{ka}{u}$
1	1	18,000	14.4	1250	0.26	0.57	2.18	540	0.40
2	1	21,000	15.2	1380	0.33	0.63	1.91	500	0.38
3	1	32,000	20.3	1570	0.36	0.71	1.96	510	0.38
4	1	24,000	17.3	1390	0.35	0.63	1.82	520	0.35
5	1	12,000	10.7	1120	0.28	0.51	1.82	510	0.36
6	1	26,000	35	740	0.37	0.34	0.91	240	0.38
7	1	15,000	36.4	410	0.37	0.19	0.50	160	0.31
8	2	23,000	24.5	1470	0.20	0.46	2.36	320	0.77
9	2	29,000	25	1720	0.21	0.54	2.61	320	0.82
10	3	......	....	....	....	....	....	...	....
11	4	2,400	10.4	230	0.29	0.17	0.57	174	0.33
12	4	4,300	15.2	283	0.32	0.20	0.64	142	0.45
13	4	4,400	18.9	233	0.29	0.17	0.59	133	0.44
14	4	2,400	8.2	292	0.28	0.21	0.75	151	0.50
15	4	4,000	11.7	342	0.30	0.25	0.82	140	0.58
16	4	6,200	19.5	318	0.34	0.23	0.68	140	0.49
17	4	2,800	9.6	292	0.30	0.21	0.72	126	0.57
18	4	5,100	18.9	270	0.30	0.21	0.66	141	0.47
19	4	6,900	25.0	276	0.38	0.20	0.53	150	0.35
20	4	2,000	8.4	238	0.31	0.17	0.57	96	0.59
21	4	5,000	18.1	276	0.30	0.20	0.66	139	0.47
22	4	6,600	24.2	273	0.33	0.20	0.59	125	0.47
23	5	31,000	4.5	7,000	0.33	....	....	...	....

[a] Assumed values. [c] Calculated values.

u = linear velocity of air through the total cross-section of the tower

$$= \frac{53{,}900}{10 \times 10} = 540 \text{ ft. per. min.}$$

$$\frac{ka}{u} = \frac{2.18}{540} = 0.0040$$

The most interesting tests are those numbered 1 to 7, which have been quoted several times in articles by different investigators. In these tests the first five were under forced draft, while the last two were under natural draft, with much reduced air velocity. A plot of ka versus air velocity for these seven tests gives a straight line passing through the origin.

It has been noticed by others[1] that the rate of cooling in towers is approximately proportional to the air velocity. Investigations at the Institute being carried on at the present time indicate that ka for humidifying apparatus in general is probably proportional to a power function of the air velocity something less than one, but for the present, at least, linear proportionality is a fairly close approximation. While k is the actual coefficient of diffusion per square foot of surface, since in many types of towers the active surface cannot be accurately measured the coefficient ka is used instead, where a represents the actual surface in square feet per cubic foot of tower. In most towers a is uniform throughout the volume of the tower.

It is therefore justifiable to call the value ka/u the "tower constant" and to use the value of this constant as a means for comparing the operation of towers of various sizes and types. Thus for tower No. 1 the average value of ka/u is 0.0037 and the average deviation from this value is 0.00021, or less than 6 per cent, while the maximum deviation is 0.0006, or 16 per cent. The important thing indicated by these tests is that ka/u was unaffected whether the tower was operated with forced or with natural draft.

Tower No. 4 has an average value of 0.00475 for ka/u, with an average deviation of 10 and a maximum deviation of 31 per cent.

Tower No. 2 has an average constant of 0.0080, which is high compared with the other towers tested. The drop in pressure through this tower was not published. It would be interesting to compare its drop in pressure with that in other towers which show smaller tower constants. It is of course obvious that a high tower constant can be obtained by obstructing the flow of air through the tower by cutting down the mean free area, but this again is at the expense of friction and back pressure. Wherever possible the drop in pressure through the tower should be measured and published.

The average value of the humid heat of air in a cooling tower is about 0.25. It will be noted that the column h/k in Table 1, which should be approximately equal[2] to s, has values of the same order of magnitude in general, but which are somewhat higher. The reason for this as predicted from the statements made in Par. 12 is shown by the fact that, while the surface of the water is at a lower temperature than the interior, making the true h for the air film greater, the value of k was calculated for the average water temperature instead of using the surface temperature. The latter is lower and would give a higher value for k. But since the vapor pressure of water rises more rapidly than the temperature, the ratio of h to k calculated on the average water temperature would be greater than that calculated on the lower surface temperature, which was the case in all but two cases in Table 1. The experimental data are therefore offered as proof of the validity of Equation [7].

Information regarding the size of tower in Test 23 was not available. It would be interesting to compare the values of ha and ka for this large tower with those of the smaller ones in the previous tests.

The results of the calculations of the test runs are felt to confirm in a remarkable manner the conceptions regarding humidification in general as developed by W. K. Lewis, furnishing as they do ample experimental proof of their validity.

Application of Equations Developed to Tower Design

The value of the foregoing equations with respect to cooling-tower design may best be shown by means of an example.

In any cooling tower the law of conservation of energy must apply. This law is represented by the heat balance of Equation [1]. Furthermore, in any cooling tower there is a necessary relationship between the amount of heat transferred by conduction and that eliminated by evaporation which is represented by Equation [8]. Finally, the capacity of any cooling tower is determined by the rate of transfer of heat and the rate of diffusion of vapor in it. These rates are dependent upon the design of the tower, and can only be determined on the basis of experimental data on the performance of towers of the type in question. The capacity factor is covered by Equation [3] or Equation [5] in the differential, but in actual design it is more satisfactory to use an integrated form of Equation [5] obtained by the use of arithmetical mean humidity differences and represented by Equation [9]. The combination of these three equations represents the conditions that must obtain in any tower and therefore serve as a proper basis for tower design.

The coefficient ka/u expresses the volumetric capacity of any particular type of tower. Experimental determination of this constant is necessary before design can be accomplished.

It is necessary for the engineer to select the type of tower most suitable for his purpose, and, from previous tests on towers of similar type, obtain values of ka which can be anticipated. In general certain specifications must be met. There may be:

Weight of water to be cooled, w (= 3000 gal. per min.)
Temperature of water to be cooled, t_1 (= 110 deg. fahr.)
Temperature to which water must be cooled, t_0 (= 80 deg. fahr.)
Average (or worst) outside air temperature, T_0 (= 80 deg. fahr.)
Average (or worst) outside air humidity, H_0 (= 0.0130, i.e., 60 per cent relative humidity)

for which the values in parentheses may be taken, those for T_0 and H_0 being the worst atmospheric conditions under which a tower must cool the water at 80 deg. fahr.

It will be noted that the condition of reducing the temperature of the water to that of the entering air is exceptionally severe and will call for a tower considerably larger than usual, since towers rarely have to meet such specifications.

There are in this case three unknown conditions, the temperature and humidity of the outgoing air, and either the volume of the tower or the ratio of the amount of air to the amount of water. The air-water ratio is usually determined by the type of tower selected and is therefore known, leaving the volume of the tower as the third unknown.

In order to calculate these unknown quantities three independent equations are needed. These may be taken as Equations [2], [8] and [9], and an integrated form of Equation [4] obtained by employing the same assumption as that used in integrating Equation [7], namely,

$$W(H_1 - H_0) = kaAx\frac{P_1 + P_0 - H_1 - H_0}{2} \quad \ldots\ldots [9]$$

Equations [2], [8], and [9] may be solved simultaneously for H_1, T_1, and Ax (which latter equals the volume of the tower). The author has seen fit to solve for W instead of Ax, but after H_1 and T_1 have been found it is easy to convert the solution for W into that for Ax.

$$W = -\frac{\dfrac{w(t_1 - t_0)}{s(2T_0 - t_1 - t_0) + r(2H_0 - P_1 - P_0)}}{1 + \dfrac{2w(t_1 - t_0)}{kaAxs(2T_0 - t_1 - t_0) + r(2H_0 - P_1 - P_0)}} \quad \ldots . [10]$$

In the particular problem in question, suppose that a forced-draft slat-type tower such as was used in Tests 1 to 7, whose tower constant ka/u is 0.0037, be selected. Towers of this type are found to be economical when handling 6.5 gal. of water per min. per sq. ft. of ground area with an air velocity of about 500 ft. per min.

The area of the proposed tower will therefore be 3000/6.5 = 460 sq. ft., and the volume of air will be 500 × 460 = 230,000 cu. ft. per min. The humid volume of one pound of dry air as read from the humidity chart is 13.8 cu. ft. Therefore—

$$W = \frac{230{,}000}{13.8} = 16{,}700 \text{ lb. per min.}$$

also—

$w = 3000 \times 8.3 = 25{,}000$ $r = 1050$ (approx.)
$t_1 = 100$ $H_0 = 0.013$
$t_0 = 80$ $P_1 = 0.0585$
$s = 0.24$ $P_0 = 0.022$
$T_0 = 80$ $ka = 0.0037 \times 500 = 1.85$

[1] Jour. Am Soc. Refrig. Engrs., vol. 3, 1916–17, p. 32.
[2] W. K. Lewis, The Evaporation of a Liquid into a Gas, MECHANICAL ENGINEERING, July, 1922, p. 445.

Substituting these values in Equation [10] and solving for Ax gives 41,500 cu. ft. for the volume of the tower, and since the ground area was determined to be 460 sq. ft., the tower height would be 41,500/460 = 90 ft. The severe conditions imposed account for the great height required.

If the tower be built as calculated, the performance under any other atmospheric conditions may be readily calculated by substitution in the proper equation.

The author realizes that the foregoing methods of calculating cooling-tower performance do not form the complete solution of the problem, and that the ultimate design of the best tower will depend upon the striking of an economic balance between the size of the tower and the cost of moving the water and the air. He feels, however, that this method of calculation is distinctly in advance of anything which has thus far appeared in print, that it furnishes a convenient and accurate tool for the designing engineer, and that it is the necessary basis for the economic balance referred to.

Finally, the author wishes to urge the inclusion of more complete and more accurate data in published accounts of tower performances. Of all the published tests studied, only those tabulated had sufficient data to enable them to be analyzed, and even then, in most cases, assumptions were necessary. Inaccurate data are often common. Test No. 10 is an example of this, the heat of vaporization alone being considerably greater than the total cooling of the water, which inaccuracy renders the test useless for purposes of analysis. A more complete knowledge of the effect of varying conditions on ha and ka can only come from studies of large numbers of accurate tests, and it is upon such increased knowledge that advance in tower design depends.

Discussion

B. H. Coffey[1] wrote that the members of the American Society of Refrigerating Engineers present at the session would remember that Mr. George Horn and he had devoted much time and study to this subject. They were therefore gratified to see the theory of cooling towers becoming of interest to the scientific men of college faculties as instanced by Mr. Robinson's paper. If the unequaled experimental facilities and mathematical ability of our great engineering schools became engaged upon this subject, they believed it could be shortly put upon a practical basis for the general profession.

All cooling towers, Mr. Coffey wrote, were more structural assemblages of cooling surface, the cost of which was by far the largest item in the installation. For this reason the designer and purchaser were peculiarly interested in the area of surface required to do the specified cooling. The discussion would therefore be confined to the points bearing on this part of the subject.

Equations [3] and [4] showed the dual heat currents that always existed between air and water when not in thermal equilibrium, both expressions containing a cooling surface term and transmission coefficient. In Equation [3] the potential or driving force was temperature difference and in Equation [4] pressure difference. For future reference they—Mr. Coffey and Mr. Horn—wished to point out that by simple transposition in each equation the surface increment would equal the heat increment multiplied by the reciprocal of the potential and consequently for zero potential the surface increment was infinite.

By ingenious use of the connecting constant s, Equation [7] was established showing the relation between the latent and sensible heat currents. The author failed to integrate this equation, however, and resorted to an approximation with which they took issue.

This approximation was based upon the assumption that the mean of the extreme differences was the true mean and the assumption was based upon the stated fact that in counter-current cooling towers the pressure and temperature differences did not change greatly between the top and bottom of the tower. No evidence was offered to support this statement, which they disputed. Referring to Table 1, tests 1 to 7, the temperature differences at the bottom of tower as a percentage of those at the top ranged from 91 to 366 per cent, and on the same basis the pressure differences varied from 83 to 148 per cent. These figures were sufficient to cast grave doubts upon the author's statement and basis of his approximation and that probably the mean potentials he obtained were not the true means.

[1] Elizabeth, N. J. Mem. Am.Soc.M.E.

If the mean potentials were incorrect the mean transmission coefficients h and k which were derived from them were incorrect and the ratio h/k would be incorrect. Referring to Table 1, col. h/k, they found this ratio varied from 0.20 to 0.38 when it should be approximately constant and about 0.25. The explanation of these wide discrepancies given by the author was not convincing to them.

The extraordinary height of tower, 90 ft., to meet the not particularly severe conditions as obtained from the simultaneous Equation [10], was they believed, due to incorrect mean potentials.

As a further test of Equation [10] they had used for final water temperature 69.7 deg. the wet-bulb temperature of the entering air. Under these conditions the total potential became zero and the cooling surface infinity, as pointed out above. The calculation resulted in a tower 230 ft. high instead of infinity, which they again would attribute to incorrect mean potentials.

In their opinion this line of attack upon the cooling-tower problem, while promising, had failed to produce working formulas that expressed the cooling process. They suggested as a basis using total potential equations. The relation $h/k = s$ gave a means of converting temperature potential into equivalent pressure potential or the reverse. The total potential could be thus expressed either in temperature or pressure alone and the problem much simplified.

They noted with regret the absence of any reference to the wet-bulb temperature in the paper, a physical quantity universally regarded of the greatest importance in this subject.

W. M. Grosvenor[1] wrote that it was a real satisfaction to find after fourteen years that a piece of one's own work stood the test of time and was still of use to other engineers, particularly in such a very admirable consideration of cooling towers as the author had given in his paper. The article on Calculations for Drier Design to which he referred had been published in the *Proceedings* of the American Institute of Chemical Engineers and the *Heating and Ventilating Magazine* for 1908, when the best available data on humidity were those of the U. S. Weather Bureau, and on these figures the calculations were based and the resulting curves plotted. The conception was there introduced of representing what might be called adiabatic evaporative cooling by lines intersecting the curves of relative humidity on a chart having temperatures as one ordinate and weight of moisture per lb. of air, volume per lb., B.t.u. per lb. of air when damp (humid heat), etc., on the other ordinate. This had proved to be a very useful and easy way of calculation. Some three years later at the annual meeting of The American Society of Mechanical Engineers for 1911, in a paper entitled Rational Psychrometric Formulae, Willis H. Carrier had used this method of presenting a newly calculated set of curves based, Mr. Grosvenor believed, on more accurate data than those of the Weather Bureau. It seemed to him that Mr. Carrier had done a very excellent and valuable piece of research work that was very thoroughly discussed at that meeting but had received too little attention since. This valuable paper contained no reference to any previous publication of humidity charts with adiabatic cooling lines, etc., but Mr. Grosvenor desired to call the author's attention to it and to ask for it the careful consideration he believed it deserved.

Now that we had in the author's work the foundation laid for a more logical and clear understanding of the data needed for perfecting the design of cooling towers, it became the obvious duty of engineers having cooling towers under operation to determine as well as they could the conditions of operation and communicate the information to him. He would then be in a position to suggest changes in operating conditions if not in design and on the basis of the results before and after could revise and perfect what should be a very valuable solution of a problem that was becoming increasingly important with municipal growth.

L. A. Phillips[2] asked if the author's formula for cooling towers which, in the example in the paper had resulted in a tower 90 ft. high, had been checked with a tower of commercial height, say, about 30 ft.

[1] Cons. Chemist and Factory Engr., New York, N. Y.
[2] New York, N. Y.

Size Selection of Dry-Vacuum Pumps

By EDWARD W. NOYES[1] and HAROLD V. STURTEVANT,[1] CLAREMONT, N. H.

In pumping from atmospheric pressure down to a given vacuum with a dry vacuum pump, as during the process of exhausting a closed tank, the volumetric efficiency of the pump varies from that at atmospheric pressure to that corresponding to the given vacuum and an average value has to be determined to use in the theoretical formulas which apply to this kind of service.

Two methods of determining the average volumetric efficiency were successively tried out by the authors and results plotted in the form of curves, but these results did not check satisfactorily with those obtained in actual tests of an installation. The process of calculation employed in the second (and somewhat more accurate) method was then reversed, and working with actual test data, values were obtained from which a "constant" curve was plotted, this curve giving closely accurate results from 80 per cent perfect vacuum up. With the aid of this "constant" curve two charts have been plotted, by means of which it is possible to determine rapidly and with sufficient accuracy (1) the size of pump required to exhaust a given volume to a specified degree of vacuum in a predetermined time; or (2), where a vacuum pump is already installed, the time required to exhaust a given volume to a specified degree of vacuum.

DRY-VACUUM PUMPS are subjected to two different kinds of service, one of which may be classified as continuous vacuum service, in which the intake and delivery pressures remain practically constant during the operation of the pump; and the other as variable vacuum service, in which the intake pressure varies from atmosphere down to some low vacuum during the operation of the pump, as would be the case in exhausting a closed tank.

The dry-vacuum pump is employed on the latter class of work in

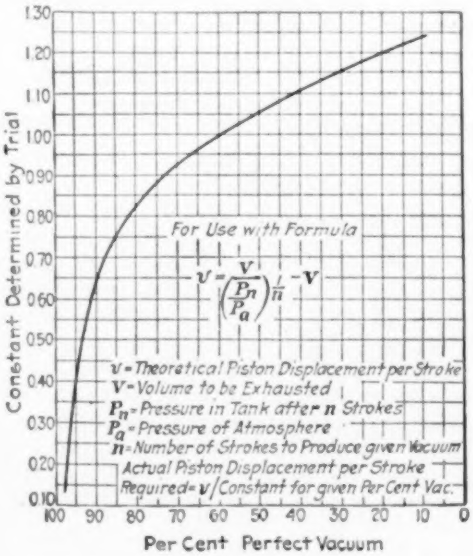

Fig. 1 Vacuum-Pump Constant for Use with General Formula for Piston Displacement

many of the industries, notably in (1) creosoting and impregnation of wood; (2) dehydrating of food products; (3) in textile and dye works to assist in the penetration of the fabrics with dyes; (4) in electrical establishments, to remove air from insulation; and (5) in the extraction of wax from waste wax paper.

The problem of determining the size of vacuum pump required to evacuate a tank of given size to a certain degree of vacuum in a predetermined time seems to be a rather elusive one, and the purpose of this paper is to present a method for its solution which will apply to any set of conditions, and which may be quickly used to determine the correct size of pump, with accuracy sufficient for all practical purposes.

[1] Sales Engineer, Sullivan Machinery Company of Claremont, N. H., and Chicago, Ill.

Presented at the Annual Meeting, New York, December 4 to 7, 1922, of The American Society of Mechanical Engineers. Abridged. All papers are subject to revision.

The experimental work described was conducted at the Claremont N. H., plant of the Sullivan Machinery Company, and the machines used were the standard, straight-line, belt-driven vacuum pumps manufactured by that company.

Two methods of determining the average volumetric efficiency were successively tried out by the authors and the results plotted in the form of curves, but these results did not check satisfactorily with those obtained in actual tests of an installation. The process of calculation employed in the second (and somewhat more accurate) method was then reversed, and, working with actual test data, values were obtained form which a "constant" curve was plotted.

Fig. 1 shows this constant curve plotted against percentage of perfect vacuum. The individual constant curves of which it is an average show some divergence between zero and 80 per cent perfect vacuum, so that within this range the results obtained by using the chart are slightly inaccurate. From 80 per cent up, however, the values given are accurate for all practical purposes.

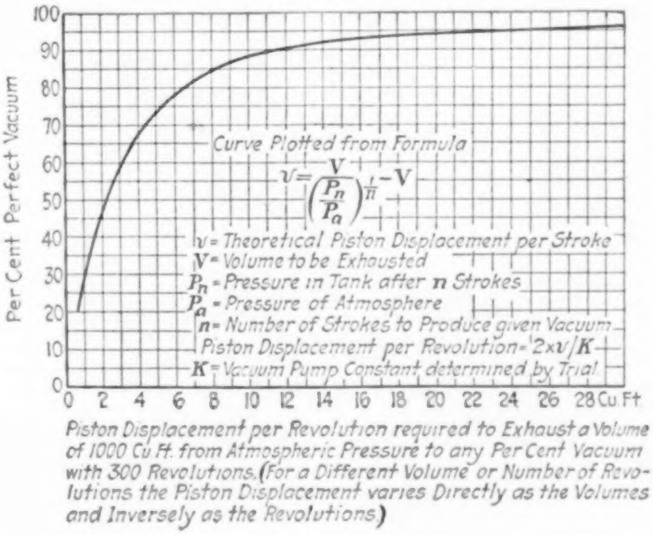

Fig. 2 Dry-Vacuum Pump Piston Displacement Required to Exhaust a Closed Space

It is evident that this curve is to be used in connection with the general formulas for solving any vacuum-pump problem rather than the average volumetric-efficiency curve, because the former has been determined from the results of tests on an actual installation.

In the general formulas given respectively in Figs. 2 and 3 there exist direct and inverse relationships between the variables which may be taken advantage of in plotting the curves. For example, in the formula of Fig. 2 v varies approximately inversely as n and directly as V, and in the formula of Fig. 3 n varies approximately directly as V and inversely as v.

This relationship makes it possible to plot the formula of Fig. 2 for some given values of V and n, such as $V = 1000$ and $n = 600$, and with v and the percentage of perfect vacuum as the variables. Then for any given case where the values of V and n are different from those in the plot, the value of v required would be directly proportional to V and inversely proportional to n. Likewise the formula of Fig. 3 can be plotted for some given values of V and v, such as $V = 100$ and $v = 0.5$, and with n and the percentage of perfect vacuum as the variables. Then for any given case the value of n required would be directly proportional to V and inversely proportional to v.

The value of v thus obtained was multiplied by 2 and divided by the constant determined by trial and plotted as piston displacement per revolution in Fig. 2. The value of n was divided by 2 and by the constant and plotted in Fig. 3 as total revolutions required.

Fig. 3 will apply to cases where the displacement of the pump is assumed or known and it is desired to find the total number of revo-

lutions required to exhaust a given volume to a predetermined vacuum. Knowing the time which may be used in the process, the r.p.m. of the pump may be obtained. A barometric pressure of 29.5 in. was used in the computations and the results plotted against percentage of perfect vacuum, so that the curve will apply with any barometric pressure. In using the curves, the average barometric pressure over a long period of time in any given locality should be used. Knowing the degree of vacuum desired, the percentage of perfect vacuum can be readily calculated. Following is a sample calculation used in the determination of this curve.

Barometer = 29.5 in.; vacuum in inches of mercury = 24; percentage of perfect vacuum = (24/29.5) × 100 = 81.3; V = 100 cu. ft.; v = 0.5 cu. ft. Then—

$P_n = (29.5 - 24) \times 0.4912 = 2.70$ lb. per sq. in.

$P_a = 29.5 \times 0.4912 = 14.49$ lb. per sq. in.

Also, from Fig. 1,

$K = 0.81$

Whence—

$$\text{Total revolutions} = \left(\frac{1}{2} \times \frac{\log 14.49 - \log 2.70}{\log (100 + 0.5) - \log 100}\right) \div 0.81 = 207.95.$$

Hence at 81.3 per cent perfect vacuum is plotted 207.95 revolutions. This same process was followed for about twenty different points and the curve thus obtained.

Fig. 2 will apply to problems in which the total revolutions or

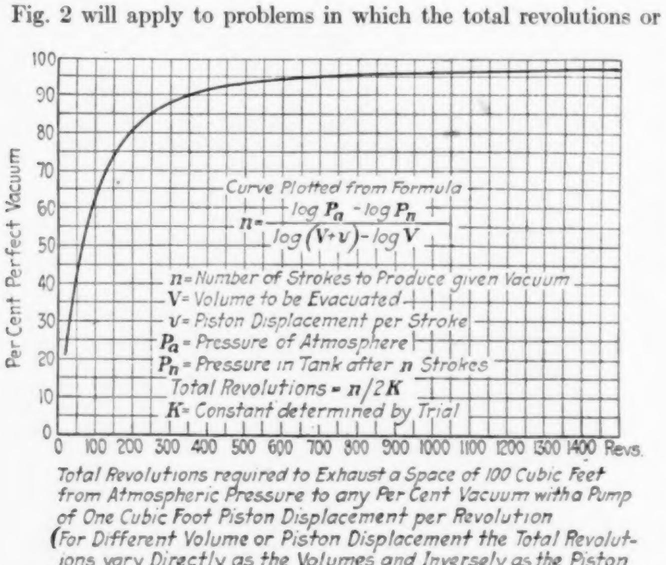

Fig. 3 Dry-Vacuum Pump Revolutions Required to Exhaust a Closed Space

r.p.m. and time are known or assumed, and it is desired to calculate the piston displacement necessary to exhaust a given volume to a predetermined vacuum. A sample calculation used in the determination of this curve follows.

Barometer = 29.5 in.; vacuum in inches of mercury = 25; percentage of perfect vacuum = (25/29.5) × 100 = 84.7; V = 1000 cu. ft. Then—

$P_n = (29.5 - 25) = 4.5$ in. $\qquad P_a = 29.5$ in.

$n = 600$

$$v = \frac{V}{\left(\frac{P_n}{P_a}\right)^{1/n}} - V = \frac{1000}{\left(\frac{4.5}{29.50}\right)^{1/0.0016667}} - 1000 = 3.138$$

Actual piston displacement per revolution = 3.138 × 2/0.76 = 8.258 cu. ft., where 0.76 is the value of K from Fig. 1 for 84.7 per cent perfect vacuum.

Thus piston displacement of 8.258 cu. ft. per revolution was plotted against 84.7 per cent perfect vacuum. This process was repeated for many different percentages of vacuum and the curve obtained.

The problems which may be solved with the aid of the formulas of Figs. 2 and 3 will vary in nature, but a few typical cases will be worked out in order to illustrate the application of the curves.

First and most important is the case where it is desired to determine the size of pump required to exhaust a given volume to a certain degree of vacuum in a predetermined time.

I—Given a volume of 5000 cu. ft., including the piping between the pump and tank, it is desired to find the size of pump required to exhaust the tank to a vacuum of 26 in n. i30 min. Barometer, 29.5 in.

a Assume that the pump will run at 280 r.p.m. In this case the percentage of perfect vacuum = 26/29.5 = 88.1. Using chart of Fig. 2, follow line at 88.1 per cent horizontally until it intersects the curve, then vertically downward to the base line, when the piston displacement will be found to be 10.15 cu. ft. This is the piston displacement per revolution required to exhaust 1000 cu. ft. from atmospheric pressure to 26 in. with 300 revolutions of the pump. Hence—

$$\text{Displacement per revolution} = \frac{10.15 \times 5000}{1000} \times \frac{300}{280 \times 30} = 1.814 \text{ cu. ft.}$$

This is the actual piston displacement required with a pump running at 280 r.p.m. to exhaust 5000 cu. ft. to 26 in. in 30 min. Allowance has been made for leaks such as are always present in the average installation. For xcessive leakage, however, a small factor of safety may be added.

b Assume piston displacement per revolution = 2 cu. ft. Using chart of Fig. 3, follow line at 88.1 per cent horizontally until it intersects the curve; hen vertically downward to the base line, where the number of revolutions is found to be 305. This is the total number of revolutions required to exhaust a volume of 100 cu. ft. from atmospheric pressure to 26 in. with a pump of 1 cu. ft. displacement per revolution. Hence—

$$\text{Total revolutions} = \frac{5000}{100} \times \frac{305}{2} = 7625$$

$$\text{R.p.m.} = \frac{\text{Total revolutions}}{\text{Time in minutes}} = \frac{7625}{30} = 254$$

This is the actual r.p.m. required with a pump of 2 cu. ft. piston displacement per revolution to exhaust 5000 cu. ft. to 26 in. in 30 min.

The second type of problem, the solution of which may be obtained by use of the curves, is the one in which a vacuum pump is already installed and it is desired to determine the time required to exhaust a given volume to a certain degree of vacuum.

II—Given an 18-in. by 8-in. pump running at 300 r.p.m., how long will it take to exhaust a tank of 2500 cu. ft. capacity from atmospheric pressure to 26 in. vacuum? Barometer, 28.4 in. Piston displacement per rev. = 2.34 cu. ft.; percentage of perfect vacuum = 26/28.4 = 91.5.

Using chart of Fig. 3, follow line of 91.5 per cent horizontally until it intersects the curve, then vertically downward to the base line, when the revolutions are found to be 415. This is the total number of revolutions required to exhaust a volume of 100 cu. ft. from atmospheric pressure to 26 in., with a pump of 1 cu. ft. piston displacement per revolution. Hence—

$$\text{Total revolutions} = \frac{2500 \times 415}{100 \times 2.34} = 4433.8$$

$$\text{Time required} = \frac{\text{Total revolutions}}{\text{r.p.m.}} = \frac{4433.8}{300} = 14.8 \text{ min.}$$

This is the actual time required, allowance having been made for leaks such as are present in the average installation.

In presenting this method of solving problems concerning the evacuation of closed spaces, the authors do not claim to have produced an absolutely accurate rule which will answer in every case. They do hold, however, that the method proposed of using a constant determined by trial gives much more accurate results than can be obtained by the use of an average volumetric efficiency computed by either of the two methods mentioned.

It will be noticed that the question of the effect of varying amounts of leakage on the size of pumps required has not been considered. This is undoubtedly an important phase of the problem and in any accurate determination of the size of pump or time required for evacuation, as the case may be, the leakage area of the system should be previously determined.

Obviously, if the system under consideration has two or three times the leakage area that was present in the system from which the curves of this paper were obtained, then the size of pump determined from them will be too small, or the number of revolutions required to exhaust the system in the given time will be too small. Yet even had this leakage factor been included, it is doubtful whether in the majority of cases it could be made use of.

Take the case of a purchase of a vacuum pump for evacuating a closed system. The chances are that the buyer has no means at hand for determining the leakage area before the pump ordered arrives, and it is unlikely that he would first install the closed system and have a test made on it before determining the size of pump required. This of course applies to an average commercial installation where the buyer, if he specifies a ten-minute period that he wishes to allow for the evacuation, will not care much if the pump provided actually requires, say, eight or twelve minutes in which to do the work.

The method presented is intended for such cases and it is assumed that leakage conditions will be approximately the same in a commercial installation as those under which the experiment described was made.

Feed Heating for High Thermal Efficiency

Economies of 25,000-kw. Power Stations Using Single- and Multiple-Stage Condenser Heaters, with And Without Economizers, Determined for the Purpose of Demonstrating The Effect of Varying the Feedwater Temperature

By LINN HELANDER,[1] EAST PITTSBURGH, PA.

For power plants using single- or multiple-stage feedwater heaters of the condenser type, the temperature of the boiler feedwater as it leaves the heaters should not be less than 150 deg. fahr. when using economizers and probably not more than 260 deg. fahr. when not using economizers, although certain conditions permit improving thermal efficiencies up to a temperature of 300 deg. fahr., corresponding to a pressure of 72 lb. per sq. in. absolute in the heater. The maximum of this range was established by consideration of fuel charges only. The lower limit of 150 deg. fahr. was chosen to avoid mechanical difficulties met when sending colder water to economizers, although this temperature, as will be seen, is less than the lowest temperature justified by purely thermal analyses of the 25,000-kw. plant used as a basis for the present studies.

That there exists for any fixed set of conditions a definite feedwater temperature at which the efficiency of power generation is a maximum is most readily demonstrated by consideration of a theoretically perfect generating plant using perfect condenser heaters. Since the feedwater heaters are considered as being perfect condensers, the pressure within them, and for the purposes of the theoretical analyses, the back pressure on the auxiliary turbine or at the extraction point on the main unit supplying the heating steam will be that corresponding to the vapor tension of the feedwater, and so will increase as the temperature of the feedwater increases. This increase in pressure in turn will increase the water rates of the steam units exhausting to the heaters, or, from another viewpoint, will decrease the work capable of being done by a pound of steam used for heating the feedwater.

Consequent upon increasing feedwater temperatures, therefore, we have a reduction in the amount of work obtainable from each pound of steam used for heating the feedwater and an increase in the total amount of steam required for heating purposes. The relation between these offsetting effects is such that the work done by the steam used for heating the feedwater increases as the temperature is increased to a certain point, after which it decreases. This is also shown by curves of Fig. 1. Any increase in feedwater temperature beyond that for which maximum work obtains continues to augment the demand for exhaust steam, but the capacity of a given quantity of the steam to do work is so reduced that the result is diminution in the total power generated by it.

Effect of Multiple-Stage Heating

A conception of what happens may be had by reference to the temperature-entropy diagrams, Figs. 2 and 3, given for both single- and double-stage heating, assuming for simplicity that saturated instead of superheated steam is used.

As the number of stages of feedwater heating increases, the work derived from the steam used for heating the feedwater increases and the temperature of the feedwater, as established for maximum theoretical efficiency, approaches that of the initial steam. Using an infinite number of stages, the temperature of the feedwater for best efficiency is equal to that of the initial steam and the efficiency of the theoretical power-generating cycle is that of Carnot's cycle.

Description of Assumed Power Station

Illustrative of what is involved in the practical problem of determining the most efficient feedwater temperatures for power stations, the effect of varying this temperature was determined for several assumed stations of 25,000 kw. capacity, using various methods of heating the feedwater. To illustrate the influence of factors such as the water rate of the main unit and the slope of the Willans line of this unit, the internal Rankine-cycle efficiency of the bled steam and the Rankine-cycle efficiencies of the house turbine, two cases—designated Case 1 and Case 2, respectively—were worked out for each arrangement of feedwater heating. The Rankine-cycle efficiency used in Case 1 for the bled steam based on the steam pressure on the turbine side of the throttle was approximately 67 per cent, slope of the Willans line was 12 lb. per kw-hr., and the water rate of the main unit when carrying the total gross station load was 10.6 lb. per kw-hr. For Case 2 the corresponding values were 80 per cent for the Rankine-cycle efficiency of the bled steam, 9 lb. per kw-hr. for the slope of the Willans line, and 10.26 lb. per kw-hr. for the water rate of the main unit.

The following are the principal data used as bases for the heat-balance study of the paper. The various formulas employed in making the study are given in an appendix to the complete paper.

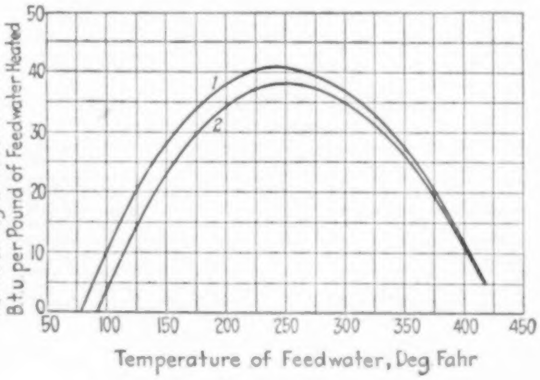

Fig. 1 Work done by Exhaust Steam

(Curve 1 shows for single-stage heating that the theoretical maximum work obtained from steam subsequently used for heating boiler feedwater is obtained with a final feedwater temperature of approximately 240 deg. fahr. when the initial temperature of the feedwater is 79 deg. fahr. Curve 2 is based on a condensate temperature of 92 deg. and shows that in this case the maximum work is obtained with the final feedwater temperature at 260 deg. Initial steam pressure, 325 lb.; superheat, 200 deg. fahr. The peaks of the curves establish the feedwater temperatures for maximum theoretical thermal efficiency. With two-stage heating and the same initial steam condition (condensate at 79 deg. fahr.) the maximum work is obtained at a feedwater temperature of approximately 300 deg., which theoretically, therefore, is the temperature for best thermal efficiency.)

Net station load	25,000 kw.
Load on auxiliary bus	1,300 kw.
Gross station load	26,300 kw.
Steam consumption of the main unit when carrying the gross station load:	Case 1: 279,000 lb. per hr. Case 2: 270,000 lb. per hr.
High-pressure drips	1000 lb. per hr.
Condensate losses	1000 lb. per hr.
High-pressure steam losses	3000 lb. per hr.
Radiation losses from low-pressure steam:	2 per cent of total heat in steam used for heating the feedwater
Pressure of steam at throttle	330 lb. per sq. in. gage
Boiler pressure	350 lb. per sq. in. gage
Superheat	200 deg. fahr.
Heat content of boiler steam	1326 B.t.u. per lb.
Heat content of steam at throttle	1324 B.t.u. per lb.
Vacuum on main unit	29 in. Hg.
Temperature of condensate	75 deg. fahr.
Temperature of make-up water entering evaporator	60 deg. fahr.
Slope of Willans line of main unit	Case 1: 12 lb. per kw-hr. Case 2: 9 lb. per kw-hr.
Heat content of the high-pressure drips recovered	390 B.t.u. per lb.
Radiation, friction and generator losses of the house turbine in kw.:	650 kw. turbine, 65 kw. 1500-kw. turbine, 135 kw.
Radiation losses from bled steam on passing through the main unit:	1 per cent of load developed by the bled steam
Internal Rankine-cycle efficiency of bled steam based on steam pressure after throttle:	Case 1: 67 per cent. Case 2: 80 per cent
Boiler efficiency when not using economizers	78 per cent
Boiler efficiency when using economizers but not including the economizer efficiency	75 per cent
Coefficient of heat transmission through economizers	5 B.t.u. per sq. ft. per deg. mean temperature difference between flue gases and water
Temperature of gases entering economizers	580 deg. fahr.
Specific heat of flue gases	0.2375
Percentage of recoverable heat recovered by economizers	85 per cent
Flue gases per lb. of coal	19 lb.
Heating value of coal	13,500 B.t.u. per lb.
Ratio of the load developed by the boilers to the full-load rating of the boilers when using economizers	2.25

The Rankine-cycle efficiencies of the house turbines are given in

[1] General engineer, Westinghouse Elec. & Mfg. Co. Jun. Am.Soc.M.E.

Contributed by the Power Division and presented at the Annual Meeting, New York, December 4 to 7, 1922, of The American Society of Mechanical Engineers. Abridged. All papers are subject to revision.

Figs. 4 and 5. An average value of 80 per cent was used for the Rankine-cycle efficiency of the bled steam on the main unit, though this will vary 2 or 3 per cent either way, depending on the conditions of bleeding and the design of the turbine.

All auxiliaries in these stations were considered as being

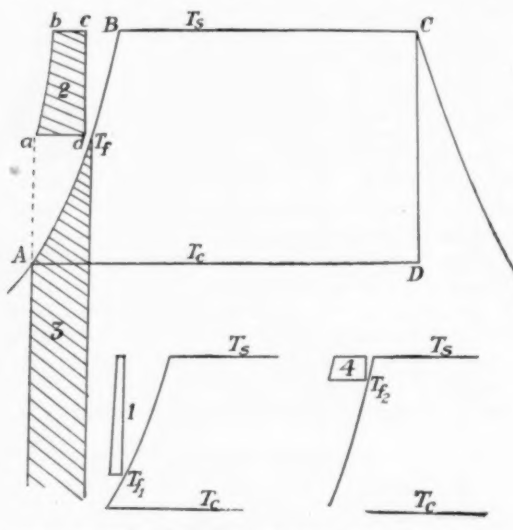

Fig. 2 Temperature-Entropy Diagram for Single-Stage Heating Using Dry Saturated Steam

(Large area $ABCD$ represents the work done by the steam passing through the main unit. The shaded area 2, above the water line, represents the work done, before entering the heater, by the steam used for heating the feedwater. The shaded area 3, below the water line, represents the heat added to the feedwater and is equivalent to the area under line ad. As indicated by areas 1 and 4 of the diagram at the bottom of the illustration, the area representing the work done by the steam used for heating the feedwater becomes rather small when the feedwater temperature deviates largely from that for best efficiency. The letters with subscripts f, f_1 and f_2 indicate the feedwater temperature for different positions of the area showing the work done by the steam usedfor heating the feedwater. The temperature of the initial steam is T_s; T_c is the temperature of the condensate.)

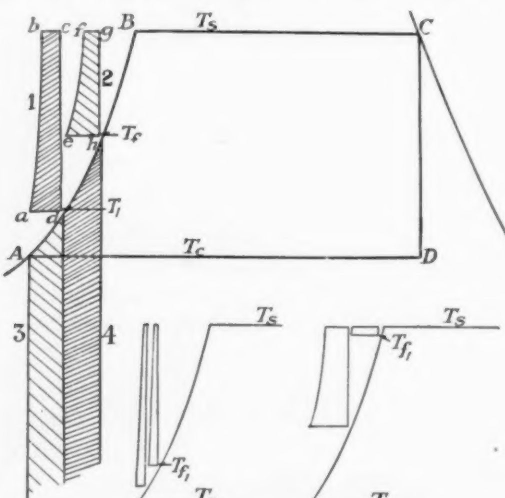

Fig. 3 Temperature-Entropy Diagram for Double-Stage Heating Using Dry Saturated Steam

(Large area $ABCD$ represents the work done by the steam passing through the main unit. Instead of a single area above the water line, as for single-stage heating, two areas, $abcd$ and $efgh$, represent the work done by the steam passing respectively to the first and second stages. For equivalent heating effects in each heater, areas 3 and 4, below the water line, are equal. The figures at the bottom of diagram show that when the feedwater temperature is rather close to either the condensate temperature or the temperature of the boiler steam, the sum of the areas representing the work done by the steam used for heating the feedwater is small. When the temperature of the feedwater equals that of the boiler steam the effect is simply that of a single-stage heater. Evidently at some temperature between that of the condensate and that of the boiler steam the sum of the areas representing the work done by the steam used for heating the feedwater is a maximum. The temperatures indicated as T_f, T_{f_1} and T_{f_2} are those of the feedwater.)

motor-driven during normal operation. Surface condensers were used on the main units and the condensate before going to the feedwater heaters was passed as cooling water through the evaporator system supplying boiler-feed make-up water so long as this operated on exhaust steam. When the temperature of this exhaust steam was too low to evaporate the water efficiently and the use of live steam became necessary, the condensate was returned directly to the feedwater heater and the water used for the condenser of the evaporator was taken from the boiler feedwater previously heated in the economizers or in the feedwater heaters. With this arrangement the use of live steam on the evaporators did not materially

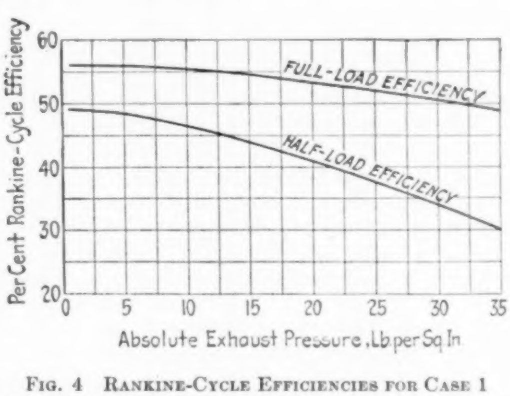

Fig. 4 Rankine-Cycle Efficiencies for Case 1

(Estimated Rankine-cycle efficiencies of 1500–1700 kw. house turbine generators as used for determining the heat balance for Case 1. The efficiencies used for the 650-kw. house turbine for Case 1 were approximately the same as those of Case 2, though the shape of its curve is similar to those given here. Steam pressure, 330 lb. gage; superheat, 200 deg. fahr.)

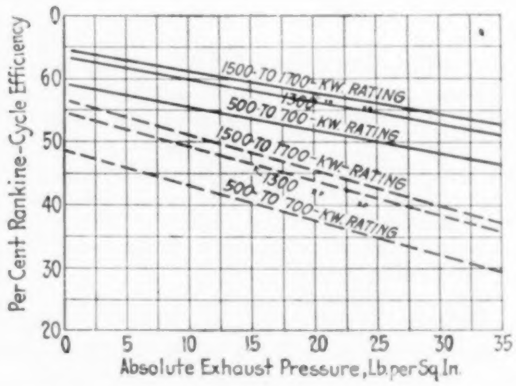

Fig. 5 Case 2. Estimated Rankine-Cycle Efficiencies of Turbine Generators Designed for Any of the Various Back Pressures and Operating against the Back Pressure for Which They Are Designed

(Steam pressure, 330 lb. gage; superheat, 200 deg. fahr. The efficiencies would deviate somewhat from those shown with back pressures below 2 lb. absolute.)

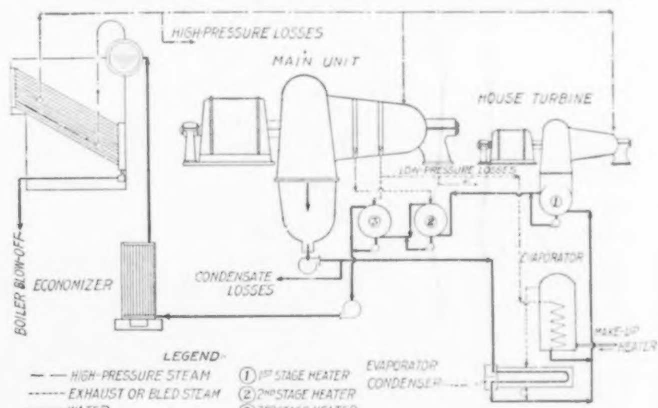

Fig. 6 Schematic Diagram of Station Layout for Three-Stage Feedwater Heating

(The condensate from the main unit is circulated through the condenser of the make-up water evaporator system before going to the feedwater heater.)

affect the heat balance, and the pressure of the exhaust steam used for heating the feedwater was determined by the pressure within the feedwater heater rather than the requirements of the evaporator system. A schematic arrangement of these plants is shown in Fig. 6. Economizers are shown, but heat balances as well were worked out for plants not using economizers, in which case the boiler feedwater was delivered directly to the boiler.

When bleeding the main unit the pressure drops in the bleeder piping varied with the amount of steam bled, and this was taken into consideration. The boilers were operated between 175 and 200 per cent of rating when economizers were not used, and under this condition had an efficiency of 78 per cent. When economizers were used the operating capacity was increased to 225 per cent of rating and the efficiency reduced to 75 per cent, not including the economizers. The power taken by the induced-draft fans was not included in the auxiliary load, and to obtain the true heat consumption for the stations using economizers the equivalent heat consumption of these fans will have to be added to the rates given.

Heat Balance for Single-Stage Heating

Heat balances for two arrangements using single-stage heating and no economizers were worked out. In one arrangement the auxiliary power was obtained from a house turbine of 650 kw. capacity, or one-half the total auxiliary load. The remainder of the auxiliary load was carried by the main unit. Steam, in addition to that supplied by the house turbine, in this case was bled from the main unit for heating the feedwater. The other arrangement used a house

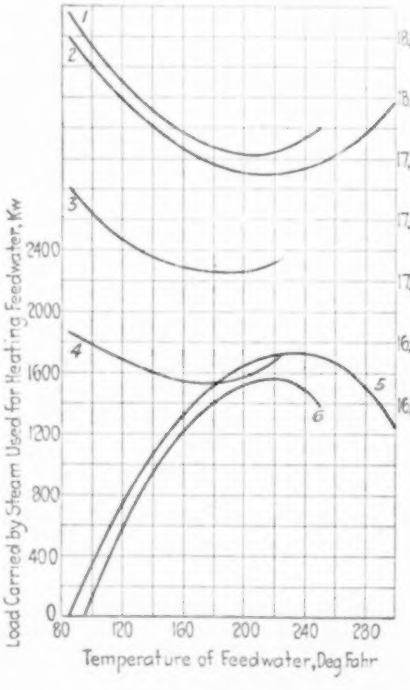

Fig. 7 Single-Stage Heating—Case 1

(Steam pressure, 330 lb. gage; superheat, 200 deg. fahr.; vacuum on the main unit, 29 in.)

Curve 1. B.t.u. per kw-hr. using a 1600-kw. house turbine and not bleeding the main unit. Economizers were not used. Feedwater temperature for best efficiency is approximately 210 deg. fahr.

Curve 2. B.t.u. per kw-hr. using a 650-kw. house turbine and bleeding the main unit. Economizers were not used.

Curve 3. B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 21,875 sq. ft. area and bleeding the main unit. Economizer surface approximately 50 per cent of the boiler area. The heat consumption of the induced draft fans, equivalent to the power consumed by them, not included.

Curve 4 B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 49,450 sq. ft. of surface and bleeding the main unit. Though the economizer area is double that for Curve 3, the feedwater temperature for best economy is reduced only 10 to 15 deg. Power taken by induced-draft fans not included with auxiliary power when determining heat-consumption curves.

Curve 5. Load developed by the steam used for heating the feedwater when using a 650-kw. house turbine and bleeding the main unit.

Curve 6. Load developed by the 1600-kw. house turbine supplying exhaust steam to the feedwater heater. A comparison of the temperature at which the peak of Curve 6 occurs with that of Curve 5 indicates the influence that decreasing Rankine-cycle efficiencies with increasing back pressures on the house turbine have on the feedwater temperature for best economy. By reference to Curves 1 and 2, showing the B.t.u. rates per kw-hr., it is seen that the real influence of the decreasing Rankine-cycle efficiencies with increasing back pressures is small.

turbine with its point of best economy at 1600 kw. for Case 1 and 1700 kw. for Case 1, and was considered as being so designed that the house turbine could deliver power to the main bus. With this arrangement no means for bleeding the main unit were provided, all steam for heating feedwater being obtained from the house turbine. As indicated by the curves 1 and 2 of Figs. 7 and 8, the first arrangement is thermally the more economical and also requires for best efficiency a slightly higher feedwater temperature than the second arrangement. It is interesting to note that whereas the theoretical feed temperature for best efficiency was in the neighborhood of 250 deg. fahr., the actual temperature is closer to 200 deg. fahr. This is due to various factors, among them being the decreasing Rankine efficiency of house turbines when operating at successively higher back pressures.

Heat Balance for Double-Stage Heating

Similarly heat balances for two arrangements using double-stage heating and no economizers were worked out on the basis that the heating effect was divided equally between the two stages. In

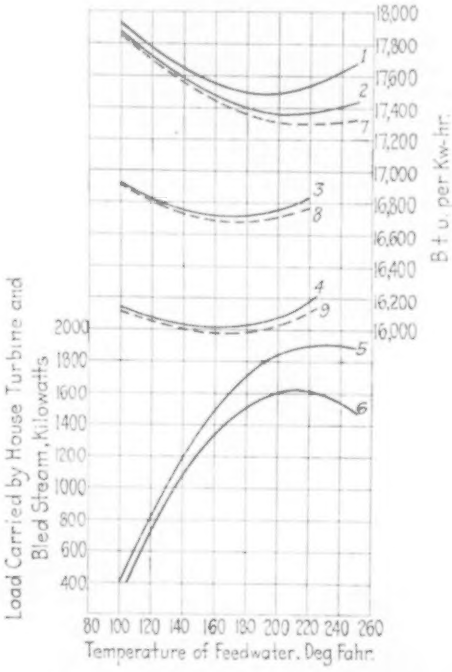

Fig. 8 Single-Stage Heating—Case 2

(Steam pressure, 330 lb. gage; superheat, 200 deg. fahr.; vacuum on the main unit, 29 in.)

Curve 1. B.t.u. per kw-hr. using a 1700-kw. house turbine and not bleeding the main unit. Economizers not used. The feedwater temperature for best economy is approximately 190 deg. fahr. This is 20 deg. lower than the best feedwater temperature for the corresponding conditions of Case 1, though due to the flatness of the curves over this range, the proper temperature for either case may be considered as approximately the same. The difference in temperature indicated is due to the difference in the slopes of the Rankine-cycle efficiency curves for the house turbine.

Curve 2. B.t.u per kw-hr. using a 650-kw. house turbine and bleeding the main unit not using economizers.

Curve 3 B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 21,875 sq. ft. area, and bleeding the main unit. The feedwater temperature for best economy is lower than the corresponding temperature for Case 1, due largely to the difference between the slopes of the Willans lines of the main units. Power consumed by induced-draft fans not included in auxiliary power.

Curve 4. B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 49,450 sq. ft. of surface, and bleeding the main unit. Power consumed by induced-draft fan was not included in auxiliary power.

Curve 5. Load developed by the steam used for heating the feedwater using a 650-kw. house turbine and bleeding the main unit.

Curve 6. Load developed by the 1700-kw. house turbine supplying exhaust steam to the feedwater heater.

Curve 7. Same as Curve 2 except that low-pressure steam losses are neglected.

Curve 8. Same as Curve 3.

Curve 9. Same as Curve 4 except that low-pressure steam losses are not included.

one of these arrangements the first-stage heaters derived the steam for heating the feedwater from a house turbine with its point of best economy at 1400 kw. for Case 1 and 1500 kw. for Case 2, only the second-stage heater deriving steam by bleeding the main unit. The other arrangement used a house turbine with its point of best economy at 650 kw., additional steam required by the first-stage heater being obtained by bleeding the main unit. The feedwater temperatures for best efficiency as indicated by curves 1 and 2 of Figs. 9 and 10 are seen to be approximately the same in either case, and about 25 deg. below that indicated by theoretical considerations alone. Both curves are rather flat over a range of 50 deg. in the vicinity of the point of best efficiency. The more efficient of the two arrangements, as with single-stage heating, is that one using the smaller house turbine, or the one bleeding the largest amount of steam from the main unit. This latter arrangement, with feedwater temperature of 275 deg. fahr., showed a possible saving of approximately 320 B.t.u. per kw-hr. as compared with single-stage heating. The maximum efficiency using single-stage heating, however, was obtained with a feedwater temperature of approximately

200 deg. fahr. Comparing single-stage with double-stage heating on the basis that this temperature of 210 deg. fahr. was not to be exceeded, the difference between double-stage and single-stage heating is about 225 B.t.u. per kw-hr.

Effect of Economizers

Economizers in connection with single- and double-stage heating were applied to those stations which used a 650-kw. house turbine and bled the main unit this arrangement being the more economical. The economizers were assumed as having a heat-recovery factor of 85 per cent and a heat-transfer rate of 5 B.t.u. per hour per deg. mean temperature difference between the flue gases and the water. Two sizes of economizers were applied to each station to illustrate the effect of varying the size of the economizers. The boilers, as previously stated, were assumed to operate at approximately 225 per cent of their rated capacity, and their efficiency in this case was taken as 75 per cent, or 3 per cent less than that used when the stations had no economizers. The temperature of the flue gases entering the economizers in all cases was taken as 580 deg. fahr., which meant that for the type of boilers selected the percentage of rated capacity developed did not vary with the temperature of the feedwater. This, of course, required that the total operating capacity be slightly decreased as the temperature of the feedwater entering the boiler was increased. The weight of the flue gases per pound of coal burned was taken as 19 lb. and independently of the feedwater temperature, but, inasmuch as the weight of coal burned per pound of steam generated varied with the feedwater temperature, the weight of gas per pound of steam generated likewise varied.

Analyses of heat balances as affected by feedwater temperatures did not involve considerations of boiler-room efficiency for those stations not using economizers. When economizers were used, however, the temperature of the flue gases leaving the economizer, for given equipment and operating conditions, was determined by the temperature of the water entering the economizer. Increasing the temperature of the water entering the economizer simultaneously increased the temperature of the flue gases so that the combined efficiency of the economizers and boilers was decreased. However, as the temperature of the feedwater leaving the heaters was increased above that of the condensate, the efficiency of converting steam to power, as previously demonstrated, increased, thereby opposing the consequent decrease in efficiency of steam generation. The relative rates at which the efficiency of steam generation decreased and that of power generation increased determined the temperature of the feedwater for best efficiency, these rates being equal for the condition of best efficiency. The rate at which the efficiency of steam generation decreases with increase in feedwater temperatures depends on the relative area of the economizers and the boilers. The larger the economizer relative to the boiler, the more rapid is the rate at which this efficiency falls off. In consequence of this, the feedwater temperatures for best efficiency were found to be lower for the larger economizers than for the smaller ones, though the differences were not of great moment.

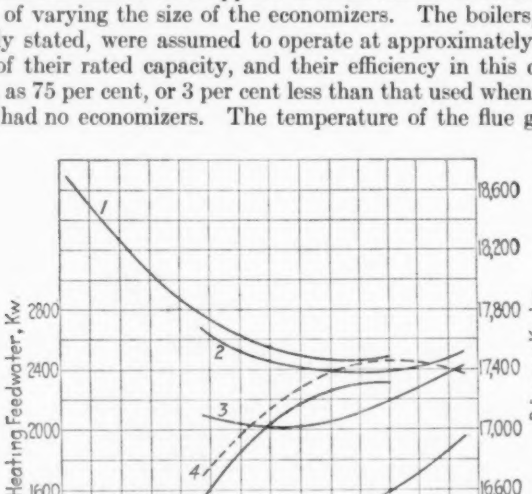

Fig. 9 Double-Stage Heating—Case 1

(Steam pressure, 330 lb. gage; superheat, 200 deg. fahr.; vacuum on the main unit, 29 in.)

Curve 1. B.t.u. per kw-hr. using a 1400-kw. house turbine and bleeding the main unit to obtain steam for the second stage, not using economizers. The feedwater temperature for best economy is approximately 280 deg. fahr.

Curve 2. B.t.u. per kw-hr. using a 650-kw. house turbine and bleeding the main unit at two points, not using economizers. It is noticeable that as the number of stages used for heating the feedwater increases, the curve of heat consumption per kw-hr. flattens out.

Curve 3. B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 18,180 sq. ft. area and bleeding the main unit at two points. Economizer surface approximately equivalent to 45 per cent of boiler area. Power taken by induced-draft fans not included in the auxiliary load.

Curve 4 (dotted). Load developed by steam used for heating feedwater when using a 650-kw. house turbine and bleeding the main unit at two points.

Curve 5. B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 47,000 sq. ft. area and bleeding the main unit at two points.

Curve 7. Load developed by the steam used for heating the feedwater when using a 1400-kw. house turbine and bleeding the main unit only for the second-stage heating. Curve 6 shows the load carried by the house turbine for this arrangement of heating the feedwater.

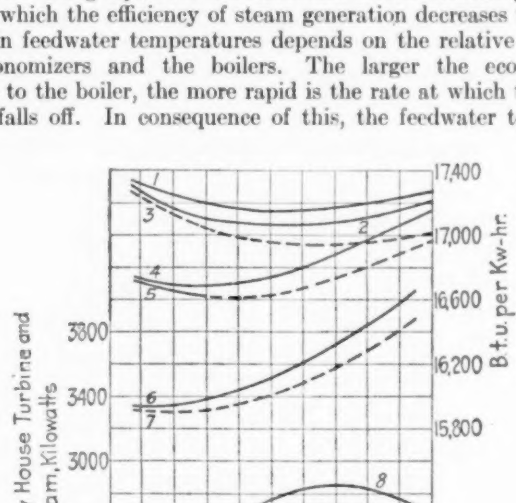

Fig. 10 Double-Stage Heating—Case 2

(Steam pressure, 330 lb. gage; superheat, 200 deg. fahr.; vacuum on the main unit, 29 in.)

Curve 1. B.t.u. per kw-hr. using a 1500-kw. house turbine and bleeding the main unit to obtain steam for the second stage, not using economizers. The feedwater temperature for best economy is approximately the same as for the same conditions of Case 1.

Curve 2. B.t.u. per kw-hr. using a 650-kw. house turbine and bleeding the main unit at two points, not using economizers. Curve 3 is the same except that losses due to leakage from the low-pressure steam piping are not included.

Curve 4. B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 18,180 sq. ft. area and bleeding the main unit at two points. The feedwater temperature for best economy is slightly lower than for Case 1. The power taken by the induced-draft fans was not included in the auxiliary power load. Curve 5 is the same but does not include the low-pressure steam losses.

Curve 6. B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 47,000 sq. ft. area and bleeding the main unit. The feedwater temperature for best economy is lower than for Case 1, due to difference in the slope of the Willans lines of the main units. Curve 7 is the same but does not include the low-pressure steam losses.

Curve 8. Load developed by the steam used for heating the feedwater when using a 650-kw. house turbine and bleeding the main unit at two points.

Curve 9. Load developed by the 1500-kw. house turbine when used to supply steam for feed heating in conjunction with bleeding the main unit.

The temperature for best economy when economizers are used, as given by the heat-consumption curves 3 and 4 of Figs. 7 and 8 for single-stage heating, lies between 175 and 190 deg. fahr. for Case 1 and 165 and 175 deg. fahr. for Case 2. With double-stage heatings, Figs. 9 and 10, the corresponding temperatures are 225 and 175 deg. fahr. for the smaller and larger economizer surfaces, respectively, for Case 2, and 230 and 200 deg. fahr. for Case 1. The temperatures for Case 1 are lower than those for Case 2, due to the difference in the slopes of the Willans lines used. For each kilowatt-hour developed by the steam used for heating the feedwater in Case 1, the steam condensed in the main unit's condenser was reduced

12 lb., while for Case 2 this figure was 9 lb. The benefit derived from carrying load on steam used for heating the feedwater was therefore relatively less for Case 2 than for Case 1, while the economizer and boiler efficiency remained the same. The differences in temperature are small, however, and indicate that with fair accuracy the desirable feedwater temperature may be considered as a range which is largely independent of the characteristics of the steam equipment used. The economizers used for double-stage heating were slightly smaller than those used for single-stage heating, commercial considerations indicating that this was justified. The difference in the sizes of the economizers did not, however, materially affect the feedwater temperature for best efficiency. The rate-of-heat-consumption curves are rather flat over a considerable range in proximity to the temperature for best economy, and increasing the area of the economizers even to the extent of doubling them need not require large changes in the temperature of the feedwater. The various data used in connection with the economizers are given in curves in the complete paper.

Comparison of Results With and Without Economizers

A comparison of feedwater temperatures for best economy as indicated by Case 1 and Case 2, respectively, shows that when economizers are not used ordinary variations in the Rankine-cycle efficiency of the bled steam, the efficiency curve for the house turbine, and the slope of the Willans line of the main unit have no considerable effect. When economizers are used, the temperatures are reduced by decreasing the slope of the Willans line of the main unit, but, as previously stated, the influence is not large. It is evident that for the purpose of establishing the proper feedwater temperature for best economy minute accuracy in the determination of the various turbine efficiencies is not required. The overall efficiency of the entire station is influenced, of course, by these efficiencies, but the best feedwater temperature changes only slightly with them. For this study a constant efficiency for bled steam was used. However, the house-turbine efficiencies were considered as being a function of the feedwater temperature, and a comparison of the temperatures for best economy obtained when using a house turbine alone with those obtained when using a smaller house turbine together with bleeding the main unit illustrates the influence of the variation in the Rankine-cycle efficiency. The difference in the temperatures that were obtained is between 10 and 15 deg. fahr., but inasmuch as the curves are flat this is of no great consequence.

Multiple-Stage Heating

By similar methods of computation, heat balances for four-stage heaters with and without the use of economizers were determined. The average Rankine-cycle efficiency for the bled steam in this case was taken at 79 per cent instead of 80 per cent, as the average efficiency over four stages would probably be somewhat less than that for one stage. Fig. 11 shows the heat consumption per kilowatt-hour for the various methods of heating the feedwater worked out for Case 2, using the 650-kw. house turbine. As the number of heating effects increase, the value of the last effect decreases, which is to be expected. It is interesting to observe also that as the number of effects increase, the heat-consumption curves for the various stations flatten out in proximity to the temperature for best economy and that it would therefore seem undesirable to go beyond a certain feedwater temperature regardless of the number of stages. When economizers of 21,850 sq. ft. are used the gains in efficiency referred to a basis of no heating for single-, double-, triple-, and quadruple-stage heating are 1.86, 2.88, 3.35, and 3.64 per cent, respectively. When the number of heating effects is increased the value of the last heating decreases, and in determining the proper number of stages consideration must be given to the investment and also to the operating problems encountered.

Air Economizers

If air for the boilers is preheated by means of air economizers and no feedwater economizers are used, such preheating does not affect the feedwater temperature for best economy. However, the air may be preheated by exhaust steam similarly to heating feedwater. Also air economizers and exhaust-steam air heating may be used with or without the coincident use of water economizers, and with single- or multiple-stage feedwater heating. If air is heated by exhaust steam alone, the problem of determining its temperature for maximum efficiency is not different from that for determining the temperature of feedwater heated only by exhaust steam. This, however, is not the case if both air and water economizers are used simultaneously.

Air and water economizers, if used simultaneously, may be placed either in parallel or in series. If placed in parallel the effect of the air economizer is to reduce the amount of gases available for heating the feedwater, and so alter the operating characteristics of the water economizers so far as these are determined by the ratio of the waste gases to the water heated. If the air heater is placed in series with the water economizer and in the coldest part of the flue gases, the leaving temperature of these gases is no longer determined by the temperature of the boiler feedwater. For such an arrangement it would appear desirable to heat the boiler feedwater to that feedwater temperature giving the maximum efficiency for the conversion of steam energy to electrical power. The feedwater would then be further heated in a water economizer while the temperature of the flue gases leaving the water economizer could be reduced to a desirable point, justified, of course, by the efficiency of the air economizers and their cost. On this basis the temperature of the feedwater entering the economizers would correspond to that previously determined when no economizers were used. The air

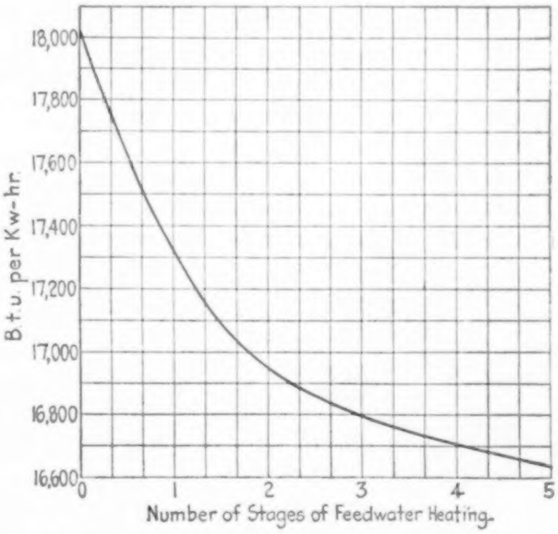

Fig. 11 Heat Consumption per Kw-Hr. for Various Stages of Feed-Water Heating—Case 2

economizers in this case, of course, would have to reduce the temperature of the flue gases below that established for the use of water economizers alone.

Conclusions

So many factors enter into the determination of the proper feedwater temperature of a plant that figures determined for one station should not be directly applied to another. As seen, the temperature should be lower for plants using single-stage heating than for those using multiple-stage heating, and also should be lower for plants using economizers than for plants not using economizers. Efficiencies of auxiliary apparatus as well as of the main generating units have their influence. If the feedwater temperature be raised much above 212 deg. fahr., factors incident to the use of pressures well above atmosphere in the auxiliary exhaust piping come into play. The proper feedwater temperature is dependent somewhat upon the initial steam pressure and temperature and on the temperature of the condensate, increasing as these increase. It is quite impossible, therefore, to determine, except within wide limits, feedwater temperatures applicable to all plants, and the present purpose has been rather to indicate in a broad way the effect of feedwater temperatures on power-plant efficiency, leaving out the matter of costs, and to give some basis for estimating the sacrifice in fuel made to assure practicable operation of the feedwater-heating system as laid out for a particular station.

Lumber Dry Kilns

By THOMAS D. PERRY,[1] GRAND RAPIDS, MICH.

The three main divisions into which lumber dry kilns may be grouped are blower, condenser, and ventilated kilns. The author, who believes that scientific kiln drying offers engineers a splendid field for research work, describes these types and discusses the possibilities of each, with particular reference to the ventilated kiln, of which there are several classes. The questions of moisture deficit and equilibrium, air interchanges, and drying cycles are adequately treated, and data showing the relation between relative and absolute humidity during the drying period are included.

THE PROBLEM presented to the manufacturer who would utilize woods intelligently is a diversified one owing to the fact that the score or more of major varieties of commercial timbers are grown in climates and seasons differing widely in temperature and moisture, and on soils ranging from the rocky hills of the Appalachians to the rich alluvial lowlands of the Mississippi delta. The origin of the lumber, has a noticeable effect on its water content. Lumber or veneer (thin lumber produced usually by rotary cutting or flat slicing, sometimes by sawing), when produced from the log, contains a large proportion of water, ranging from 25 to 75 per cent of the total weight. One square foot (board measure, one inch thick) of gum lumber, weighing approximately five pounds when sawed, will be reduced to about three pounds when its water content of approximately one quart has been evaporated. Oak grown on a hillside may contain only a pint (approximately 1 lb.) and swamp gum may have from 2 to 4 pints of water per sq. ft. B. M.

This water content of wood exists in two forms, free moisture and cell moisture, the former being readily evaporable in ordinary air drying, and the latter demanding excessive air drying (several years) or artificial treatment in kilns. The usual border line between the two forms of moisture is in the vicinity of 30 per cent moisture content (the percentage of the weight of water removed being computed on the dry fiber weight as a base). It is possible to use artificial means to remove this free moisture, but a simple air exposure is usually more economical.

By far the largest volume of lumber products are dried in the form of lumber or veneer. The drying of unusual forms and shapes, such as staves, handles, shingles, laths, etc., is decidedly specialized and outside the range of this discussion.

Types of Kilns

The original artificial drier was a smoke kiln, now practically obsolete. This was followed by the furnace kiln and the steam-coil kiln. Some woodworkers still persist in using home-made equipment, but kiln design and building has become a special branch of manufacturing that has its own recognized field.

The three main divisions into which lumber kilns may be grouped, as illustrated in Fig. 1, are:

1 Blower: Mechanically forced ventilation, or recirculation, whether suction or plenum method; moisture-laden air usually discharged out of doors

2 Condenser: Generally of the gravity recirculating type, in which the air passes over moisture-removal or condensing units, once in each interchange

3 Ventilated: Fresh air taken in and used air discharged direct from kiln to atmosphere, utilizing the fundamental laws of physics to obtain internal circulation.

To understand the particular range of each of the foregoing types it is necessary to outline the moisture-removal problem in some detail from the standpoint of the lumber to be dried. Air drying removes more moisture from the surface than from the center, and owing to the length and width of a board, drying takes place chiefly through the flat faces, rather than through the ends or edges. The inevitable result is a surface drier than the interior, and air-dried stock is therefore subject to an internal strain that often manifests itself in the form of warp, twist, or other surface irregularities. The problem is to draw the center moisture out and have surface and interior equally dry.

The skill of an engineer is not required to discover that, if the drying of wood afforded no organic difficulties, the blower would dry rapidly, utilizing considerable power, and the condenser would function slowly but at low cost.

The ventilated kiln is the least understood, even though most generally used, and offers an unusual opportunity for speed and efficiency when its underlying principles are grasped.

The nature of wood is the phase of the problem that engineers least appreciate. It is almost axiomatic that wood should not be subject to external or internal strain during drying, but it is practically impossible to obtain such a "strainless" condition. The reduction of this drying strain to a minimum point is necessary to drying without damage to the lumber, the usual manifestations of which are checking, warping, honeycombing, "hollowhorning," etc. The chance of internal strain greatly increases with the thickness of the wood to be dried: e.g., $^1/_{16}$-in. veneer is practically all surface and can develop little internal strain in any kind of drying, while 4-in. green oak presents a decidedly stubborn drying problem, and taxes the skill of the best operator in an efficiently controlled kiln.

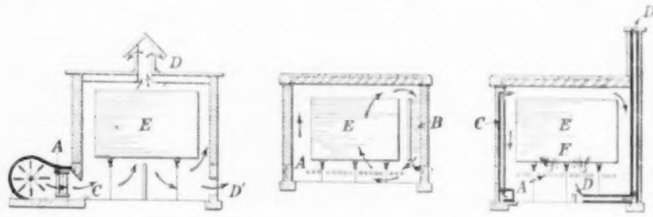

Fig. 1 Showing (from Left to Right) Blower, Condenser and Ventilated Kilns

(*A*, Heating unit; *B*, condensing unit; *C*, fresh-air inlet; *D* and *D'*, used-air outlets; *E*, lumber charge; and *F*, humidifying unit.)

Blowers and Condensers

These facts lead to the logical conclusion that where thickness is nominal, as in the case of veneer, applied shellac, stain, filler, varnish, and glue, the speed of the air movement in a blower kiln will remove moisture rapidly and cause no serious damage; but where thickness becomes appreciable ($^1/_2$ in. and up) the rapidly moving atmosphere of a blower kiln will produce uneven drying and unnecessary interstrain that will inevitably damage the lumber. Even with attempted maintenance of high humidities (difficult in blower kilns[1]) the hazard is serious. On the other hand, the slowness of the condensing kiln will make for accurate control, but the lack of speed may force the initial cost of installation to an excessive amount. As a matter of fact, Government experts have frequently recommended the condensing kilns without comprehending in most cases the economic aspect. It has been the author's experience during the last five years that kilns of the condensing type used for war work did no better drying than the best of the ventilated types, and required for an equivalent output practically twice as many units, at approximately double the initial cost per unit, or a quadruple investment, with no appreciable gain in quality or safety.

As a conclusion it may be noted that blower kilns are most suitable where thickness is not a factor, and that condenser kilns are especially serviceable in the drying of thick green woods where internal strain is a decided danger.

Typical Drying Cycle

Before considering the individual characteristics of the various types of ventilated kilns, it may be well to outline briefly a drying cycle. Take an actual operation schedule, tested by practice,

[1] Vice-president and manager, Grand Rapids Veneer Works. Assoc-Mem. Am.Soc.M.E.

Contributed by the Forest Products Division for presentation at the Annual Meeting, New York, December 4 to 7, 1922, of The American Society of Mechanical Engineers. All papers are subject to revision.

[1] Kent's Mechanical Engineers' Pocket Book, 1916 edition, p. 573.

of reducing 1-in. oak from 35 to 5 per cent moisture content in 16 days of 24 hours steam. Plotting this schedule, curves showing the variations in temperature and the relative humidity during the period of drying are obtained as shown in Fig. 2. The drying cycle divides itself into three phases: initial "steaming" or high-humidity period in which lumber is heated through and enough moisture added to make the surface as wet as the center; intermediate "stewing" or cooking,[1] a transition period; and the final "drying," when humidity may be dropped and temperature raised within limits demanded by the kind and condition of lumber.

The temperature should gradually rise from 125 to 170 deg. fahr. and care should be exercised that it does not go up too fast during the initial steaming, which would tend to crust the outside of the boards. The relative-humidity line reaches 100 per cent as rapidly as possible without producing an excess of temperature. The reason for the high initial humidity is that partly dried lumber, as placed in the kiln for drying, normally has a surface drier than the interior, and unless the surface is thoroughly moistened the internal moisture will be sealed in by the dry and shrunken surface layers; in other words, the degree of internal strain will be intensified rather than

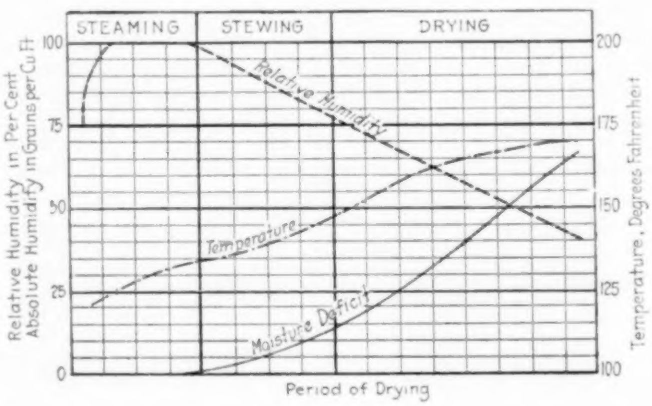

FIG. 2 TEMPERATURE AND RELATIVE HUMIDITY DURING DRYING CYCLE AND CURVE SHOWING MOISTURE DEFICIT

reduced. If the surface of the board is steamed or exposed to an atmosphere of high humidity it will absorb moisture and expand, making it possible for the cellular structure to conduct additional moisture to the surface, and reducing the internal strain. Heating the lumber through is also accomplished quickly and safely by this steaming.

Moisture will pass out more rapidly from the surface of a board the center and surface of which are approximately equal in moisture content, and the final result of the drying cycle will be a reasonably uniform moisture content from face to face of the board and an absence of internal strain, with consequent damage. It is a rather significant fact that lumber dried by the so-called "wet" process explained in the preceding paragraph is more plump than when treated by the old-fashioned method of "giving it all the heat it will stand" without preliminary steaming or due regard to the humidity. Experiments have proved that the endeavor to dry without steaming takes longer because moisture transfuses more slowly through a dry than a wet surface, and the resulting damage is usually serious.

RELATIVE VS. ABSOLUTE HUMIDITY—MOISTURE DEFICIT

Another way of expressing the function of humidity is to express the cycle of drying from the standpoint of absolute humidity, or the grains of water vapor per cubic foot of atmospheric kiln content. Table 1 shows the relation of absolute (grains of water per cubic foot) and relative humidity (percentage of saturation) within the usual ranges of kiln temperature.

If we could isolate and examine a cubic foot of space (containing air and water vapor) in a kiln at a temperature of 150 deg. fahr. and a relative humidity of 50 per cent, we would find that it contains 36.76 grains of water in the form of vapor and possesses the capacity, before reaching saturation, of absorbing an equal additional amount of water vapor. It is this difference between actual grainage and the grainage of saturated water-vapor content that expresses the drying power of the kiln.[2] Stated in another way, this means that a cubic foot partly saturated has a tendency to become entirely saturated if free moisture is accessible, i.e., from the lumber.

This moisture deficit may be termed the measure of drying power in a kiln (when considered in connection with the temperature and circulation) and the greater the deficit the greater will be the pull exerted on the moisture contained in the lumber. At the beginning of the drying operation, when reducing the lumber to a uniform wetness, the moisture deficit is nil; and toward the end of the operation, at a temperature of 170 deg. fahr. and a relative humidity of 40 per cent, each cubic foot is eagerly seeking for 67 grains of water vapor or such part of it as can be extracted from the lumber. Fig. 2 contains a curve showing the moisture deficit expressed in grains per cubic foot. It will be noted that this line curves up more rapidly than the temperature line. This moisture deficit may be termed the measure of potential drying power. It is easy to remove moisture from the lumber at first, but the last few stages of drying require a decidedly strong pull.

While it would be possible to reduce the relative humidity to below 40 per cent (carrying the moisture deficit above 67 grains), it would result in too rapid a rate of surface drying and leave an unevenly dried product. The secret of controlling temperature and humidity in efficient drying is to avoid any sudden changes during the progress of the operation and to finish without carrying the temperature to an extreme that will damage the wood; and to avoid reducing the relative humidity at any point where it will dry the surface faster than the center.

AIR INTERCHANGES

The number of air interchanges per hour should not be too great. If an anemometer test shows three or four complete air changes per hour, it is enough. The process of the withdrawal of water

TABLE 1 GRAINS OF WATER VAPOR PER CUBIC FOOT AT VARIOUS RELATIVE HUMIDITIES[3]

(7000 grains = 1 lb. avoirdupois)

Temperature, Deg. Fahr.	Relative Humidity 5%	10%	15%	20%	25%	30%	35%	40%	45%	50%	55%	60%	65%	70%	75%	80%	85%	90%	95%	100%
100	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
105	1	2	3	5	6	7	8	9	10	11	13	14	15	16	17	18	19	21	22	23
110	1	3	4	5	7	8	9	10	12	13	14	16	17	18	20	21	22	24	25	26
115	2	3	5	6	8	9	11	12	14	15	17	18	20	21	23	24	26	27	29	30
120	2	3	5	7	9	10	12	14	15	17	19	20	22	24	26	27	29	31	32	34
125	2	4	6	8	10	12	14	16	18	20	22	23	25	27	29	31	23	35	37	39
130	2	4	7	9	11	13	15	18	20	22	24	26	29	31	33	35	37	40	42	44
135	3	5	8	10	13	15	18	20	23	25	28	30	33	35	38	40	43	45	48	50
140	3	6	8	11	14	17	20	23	25	28	31	34	37	39	42	45	48	51	54	56
145	3	6	10	13	16	19	22	26	29	32	35	38	42	45	48	51	54	58	61	64
150	4	7	11	14	18	21	25	29	32	36	39	43	46	50	54	57	61	64	68	72
155	4	8	12	16	20	24	28	32	36	40	44	48	52	56	60	65	69	73	77	81
160	4	9	13	18	22	27	31	36	40	45	49	54	58	63	67	72	76	81	85	90
165	5	10	15	20	25	30	35	40	45	50	55	60	65	71	76	81	86	91	96	101
170	6	11	17	22	28	34	39	45	50	56	61	67	73	78	84	89	95	101	106	112
175	6	12	19	25	31	37	44	50	56	62	69	75	81	87	94	100	106	113	119	125
180	7	14	21	28	34	41	48	55	62	70	76	83	90	97	103	110	117	124	131	138
185	8	15	23	31	38	46	54	61	69	77	84	92	100	107	115	123	130	138	146	153
190	8	17	25	34	42	51	59	68	76	84	93	101	110	118	127	135	144	152	160	169
195	9	19	28	37	47	56	65	75	84	94	103	112	122	131	140	150	159	168	178	187
200	10	21	31	41	51	62	72	82	92	103	113	123	133	144	154	164	175	185	195	205

[1] "Stewing" used in the sense of a vapor bath, rather than as cooking (212 deg. fahr). Some authorities merge this into the "drying period."
[2] It is obvious that the higher the temperature, with a given moisture deficit, the more rapid the evaporation.
[3] Adapted from tables compiled by Dr. William M. Grosvenor, with intermediate readings interpolated and decimals discarded.

vapor from the interstices of the lumber cannot be very rapid. Too rapid air movement in a gravity kiln subjects it to the same criticism as a blower kiln, i.e., too rapid surface drying.

High humidity and large circulation of air are antithetical to one another. To obtain high humidity, the circulation must be either stopped altogether or greatly reduced, and to reduce the humidity a greater circulation must be induced by increasing the draft.

The following example, based on three interchanges per hour, illustrates this point:

Size of kiln inside	15 ft. high by 19 ft. wide by 26 ft. long
Cubic contents, gross	7410 cu. ft.
Lumber allowance	1083 cu. ft. (13,000 ft. B. M.)
Equipment allowance	50 cu. ft.
Cubic contents, net	6277 cu. ft.

Allowing a moisture-lifting capacity averaging 25 grains of water vapor per cu. ft. of each discharge from the kiln:

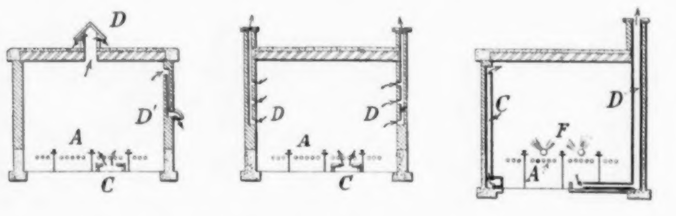

Fig. 3 Classes of Ventilated Kilns

(From left to right, ventilated at top, side and bottom. *A*, heating unit; *C*, fresh-air inlet; *D* and *D'*, used-air outlets; and *F*, humidifying unit.)

Discharge per cu. ft.	25 grains or 0.00357 lb. avoirdupois
Total single discharge	0.00357 × 6277 = 22.42 lb.
Hourly discharge (3)	67.26 lb.
Daily discharge (72)	1614.25 lb.

Estimating the average water content of saturated oak lumber at 1 lb. per ft. B. M.:

Lumber allowance	13,000 ft. B. M.
Moisture to be removed	13,000 lb.
Time required	13,000 ÷ 1614 = 8.05 days

This would indicate that the actual drying period (Fig. 2) will require practically eight days if the texture of the lumber permits of an average removal of 25 grains. An additional time allowance should be made for steaming.

The removal of 2 lb. moisture per ft. from green gum, for instance, would necessitate either increasing the drying time proportionately, or operating at a higher temperature.

Moisture Equilibrium

Theoretically it may be possible to reduce lumber to absolute dryness, but it cannot be kept so except in sealed receptacles. Under ordinary factory-workroom conditions, it will reabsorb to 6 or 8 per cent moisture. It is necessary, therefore, only to dry down below normal and allow for reabsorption. In ordinary lumber yards the lumber will rarely air-dry below 15 per cent, and kiln-dried stock left out of doors will absorb to about the same point. For structural purposes, exposed to weather the moisture content of timbers should be around 15 per cent to prevent shrinkage or expansion. For more accurate work the equilibrium point should be ascertained for each plant and will be found for interior trim to be about 8 to 10 per cent and for furniture, pianos, etc., from 4 to 7 per cent.

Ventilated Kilns

Ventilated kilns are readily grouped into three classes as indicated in Fig. 3.

Kilns of the first class are ventilated at the *top*, either through the ceiling and a cupola on the roof, or through openings in the side walls near the ceiling. Fresh air is usually admitted near the floor and the air movement must be distinctly upward. The real problem is to obtain an efficient utilization of this air movement to lift moisture out and up from the lumber.

No difficulty will be encountered in securing ample air discharge, as hot air will always rise and seek an exit. It is obvious that the hot air escaping will not be heavily charged with moisture. Only to a limited extent is it possible to use this upward-moving air as a moisture vehicle, because the increased water-absorption capacity accompanying higher temperatures will carry only a small amount of water vapor without a net increase in weight.

As an example, suppose the temperature of the air in the lower part of the kiln is 140 deg. fahr. and at the top 160 deg. fahr.; dry-air weights, 463.09 and 448.11 grains per cu. ft., respectively, a difference of 14.98 grains (less than 20 per cent relative humidity at 160 deg.). Air, therefore, cannot move upward and carry as much as 15 grains (neglecting expansion which is less than 4 per cent). Compare this with the fact that in a down-draft kiln the air at 140 deg. can carry 56 grains per cu. ft. before reaching saturation.

Any steam spray in this type tends to monopolize the limited moisture deficit of the kiln air and to greatly decrease the drying power as the air circulates upward through the lumber.

The second class have air exits at the *sides* and air inlets usually at the bottom. There is less of a scientific basis for this type than for either the first or the third type. There is great danger of "short-circuiting" the air across a corner of the kiln, thus causing pockets in places where a lack of air movement prevents drying. When this type is applied to the primary drying (to reduce shipping weight) at sawmills, with high temperatures, it may give fair results in regard to the removal of free moisture, and perhaps will reduce the moisture content to 15 per cent, but below this point it is undependable.

The use of steam spray or the handling of relatively moisture-laden atmosphere is no more possible than in the first

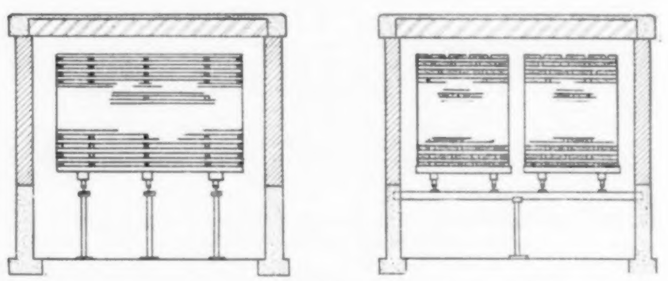

Fig. 4 Transverse Section of a Kiln, Showing Cross-Pile (at Left) and End-Pile (at Right)

(Notice somewhat better air circulation opportunity in end-pile.)

type. Neither of these types are adapted to what might be termed final drying for musical instruments, furniture, high-grade cabinet work, interior trim, or any object where permanence of dimension and a high degree of finish are required.

The third type, which is often called "reversed" ventilation, has a downward circulation with outlets at the *bottom*. The fresh-air inlets are at or near the ceiling line and the damp-air exits are in the pit below the lumber. This method adapts itself most effectively to the handling of heavy moisture-laden air shown to be necessary to dry the lumber uniformly. Whenever any portion of the atmosphere of the kiln acquires a substantial amount of moisture, it will lose temperature, increase in weight, and settle. It is therefore necessary to provide an adequate damp-air accumulation pit below the lumber, and to supply chimneys or stacks with sufficient draft to carry this moisture-laden air up and out. Heavy air that has settled into a pit will not readily be drawn up a stack. Steam-heated pipes in the stacks will accomplish this and at the same time keep the water vapor from condensing on the interior of stacks or at their top outlets. This type gives opportunity for excellent control, as any humidity can be handled and the air discharge be under positive control under various climatic conditions. For high-grade work it affords reliable regulation of the three essentials: temperature, humidity, and circulation.

It has been the purpose of this paper to outline some of the fundamental principles in lumber drying and to indicate their most important applications. Many of the combinations of mechanical apparatus are adequately covered by United States and foreign patents, although the fundamental elements have been in use too long to have further protection.

(*Continued on page 120*)

Diesel-Engine Clutch Used in the German Submarine "U-117"

By W. H. Nicholson,[1] Camden, N. J.

The purpose of this paper is to give to American builders of Diesel engines the benefit of the author's knowledge of German Diesel-engine clutches obtained at the time of the dismantling of ex-German submarines in the United States. The U-117's clutch is described in detail and other types of German Diesel-engine clutches are covered generally.

The submarine *U-117* was commissioned by the Germans in 1917 and operated off our North Atlantic coast in that summer and in the summer of 1918. It is known that this submarine took part in the sinking of coal barges and fishing boats in the vicinity of Nantucket and it is supposed that she is responsible for the planting of mines off the Long Island Shore, one of which sank the U. S. Cruiser *San Diego*.[2]

The *U-117* is a large mine-laying and -operating submarine. Her overall dimensions are: length, 275 ft.; beam, 17 ft.; and draft, 10 ft. She has a carrying capacity of 20 torpedoes and 45 mines. The armament consists of a 6-in. rapid-fire gun forward

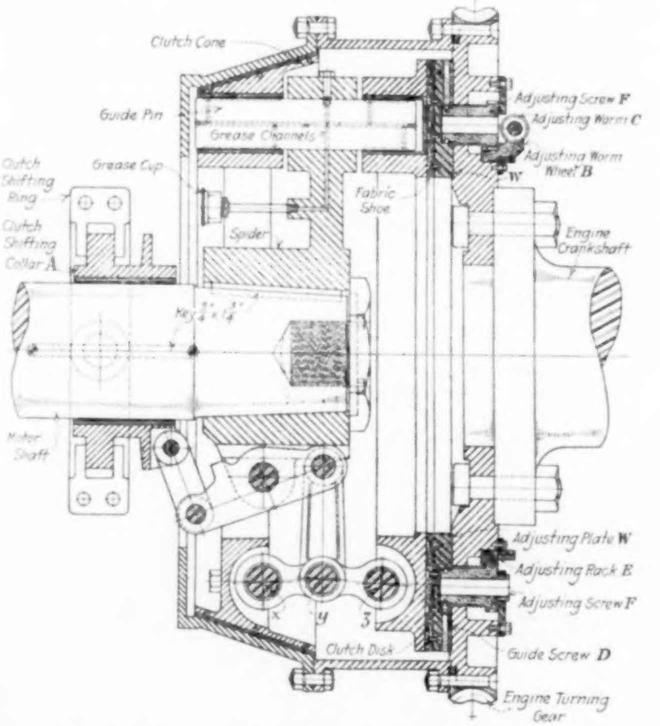

Fig. 1 Sectional View of Diesel-Engine Clutch Used in the German Submarine *U-117*

and a 5-in. gun aft, both on the main deck. There are four torpedo tubes in the compartment forward, and two mine tubes in the compartment aft.

The engine equipment consists of a port and a starboard Diesel oil engine, four-cycle, six-cylinder with two air-compressor cylinders of 1200 i.hp. at 450 r.p.m. for surface cruising and battery charging. The bore of the working cylinder is 17.73 in. and the stroke 16.5 in. The complete engines weigh 57,000 lb. each.

There are also the port and starboard main motors, each rated at 600 hp. at 332 r.p.m., for submerged operation. In general, the design of the motor comprises two compound interpole assemblies in one frame with the armatures in tandem on the same shaft motor-generator fashion. The motors receive their power from two 124-cell 248-volt storage batteries. When the boat is under way on the surface, each motor unit is driven as a generator by the Diesel oil engines, using the excess power of the engine to recharge the batteries and carry the auxiliary power load.

The log of the *U-117*, her general description, and the description of her propelling machinery give some idea of the work performed by her clutches.

Details of the *U-117* Main Engine Clutch

The clutch is located between the Diesel oil engine and the main

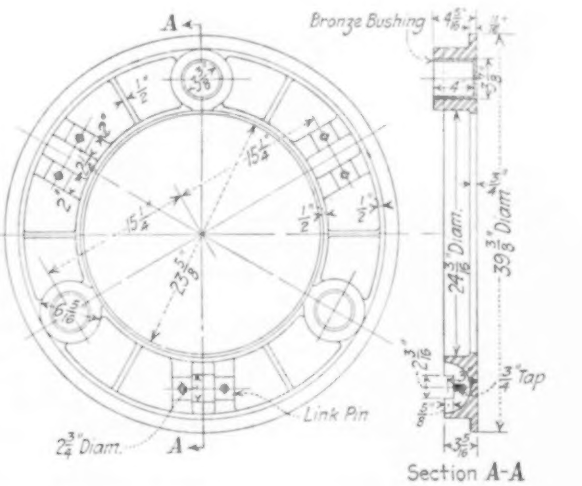

Fig. 2 Clutch Disk

motor, both port and starboard. It is of the single cone and disk friction dry type, constructed of semi-steel and cast iron, and operated by compressed air. The clutch was built to metric measure. All dimensions given in this paper, however, are in nearest inches or fractions of an inch.

The outer casing, adjusting plate, worm wheel, and adjusting screws, which serve as a flywheel, are carried on the engine crankshaft. The casing as shown in Fig. 1 is made of two parts bolted together. The inner surface of the after casing is machined to a cone surface; the forward casing is a straight sleeve or spacer for power transmission only. The extreme outside diameter of the casing is 45 in.

The inner clutch cone or male cone and clutch disk are carried on the guide pins of the spider, on which they are a sliding fit. They are of semi-steel and the outer surface of the male cone is finished to conform to the conical surface of the female or flywheel part. The forward surface of the clutch disk is finished for contact on the fabric shoe of the adjusting plate. Figs. 2 and 3 are detail sketches of the male cone and the clutch disk carried by the spider. The taper of the cone surfaces is about 14 deg. The cone and disk are bronze-bushed for the guide pins.

To the outer surface or periphery of the male cone there is attached, by means of countersunk machine screws, a fiber shoe as shown in Figs. 1 and 3. The shoe is $^1/_4$ in. thick and is made up in four sections, the divisions being in the direction of the axis of the guide pins or shaft. It is the surface of the fiber shoe that contacts with the metal surface of the female or outer casing when the clutch is thrown in.

The spider, attached to the motor shaft by a single key and a retaining screw has three guide pins parallel to the axis of the engine and motor shafts and extending through the spider on either side

[1] Designer, Engr. Dept., Westinghouse Elec. & Mfg. Co.

[2] The *U-117* was recently used as a target and sunk off the Virginia Capes in the U. S. Army airplane bombing experiments.

Contributed by the Gas Power Division for presentation at the Annual Meeting, New York, December 4 to 7, 1922, of The American Society of Mechanical Engineers. All papers are subject to revision.

as shown in Figs. 1 and 4. The guide pins are steel forgings and are shrunk in the spider.

The clutch is thrown in by forcing the male cone and the disk apart until the male and female cone surfaces engage and the disk surface engages the fabric shoe of the adjusting plate; and by drawing the male cone and the disk together and thus releasing the surfaces the clutch is thrown out. The male cone and the

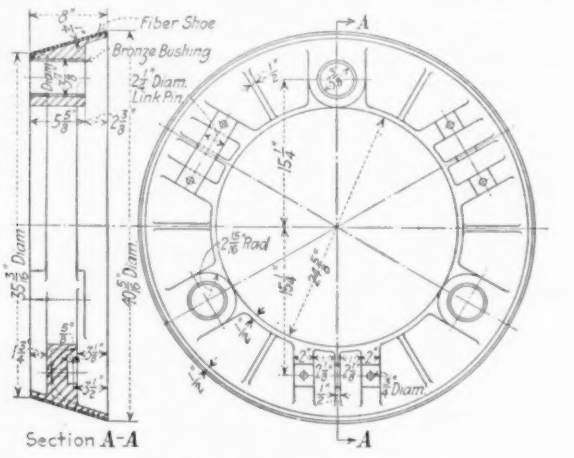

Fig. 3 Clutch Cone

disk are drawn together or forced apart by a series of links and toggles actuated by a sleeve or collar (A in Fig. 1) sliding on the motor shaft. The shifting collar is shown in detail in Fig. 5. It is babbitted for a sliding fit on the motor shaft and is single-keyed thereto.

Referring to the lower half of Fig. 1, the center x is shown fixed to the male cone and center z to the disk. The clutch is engaged by separating centers x and z and by forcing center y up to a line adjoining centers x and z. There are three sets of operating links and toggles equally spaced between guide pins of the spider as shown in Fig. 4.

To the after side of the worm wheel there is attached by means of six adjusting screws a steel disk or adjusting plate faced with fabric. The fabric is $^7/_{16}$ in. thick and serves the same purpose as the fiber shoe of the male cone. The fabric is also in four sections and is secured by means of countersunk machine screws. The sectional divisions of the fabric are radial. When the clutch is engaged, the forward surface or finished surface of the clutch disk has contact with the fabric shoe.

The fore-and-aft movement of the male cone and disk is $^3/_8$ in., or a total of $^3/_4$ in. The fiber and fabric shoes have some flexibility and it is these surfaces that absorb the load shocks and transmit the power of the engine. The design of this clutch differs in this respect from the clutch used with the 3000- and 1750-hp. engines, in that those clutches have, when engaged, a metal-to-metal contact on lubricated surfaces. The fabric and fiber shoes may be renewed when worn.

The adjusting worm wheel is carried on the engine turning gear (see B in Fig. 1) and is rotated by the adjusting worm (C) which is hand-operated from either end of the worm shaft by means of a wrench. The adjusting worm is bracketed to the engine turning-gear cover plate. Any movement of the adjusting worm is transmitted through the rack (E) and rotates the adjusting screws (F) which are threaded through the worm wheel. There are six adjusting screws equally spaced, three of which are secured to the adjusting plate by means of the guide screws (D). The other three adjusting screws have contact only on the forward surface of the adjusting plate; their ends are flat and do not penetrate the plate. The sketches do not show the short adjusting screws. By means of the adjusting mechanism there is a movement of $^9/_{16}$ in. between the disk and the fabric shoe, the adjusting plate moving aft from the turning gear that distance at the surfaces marked W in Fig. 1.

The guide pins of the spider are lubricated by means of three grease cups attached to its arms by a length of pipe. The guide pins and spider arms are drilled for the passage of grease and the former are grooved for its distribution. Plugs are inserted in the blind ends of drilled holes. The circumferential force of centrifugal force of the clutch should carry sufficient grease to the pins without attention to the grease cups except for occasional filling.

The clutch shifting cylinder shown in Fig. 6 is of semi-steel. The bore is 7 in. and the stroke is from $7^1/_2$ to 8 in., depending on the setting of the pistons. The pistons may be set for long or short stroke by means of the piston adjusting nuts. To prevent pistons from striking cylinder heads when operating, adjustable stops are provided on the guide rods and spring stops are secured to the cylinder heads. The springs at either end of the cylinder absorb the shocks when throwing the clutch in or out.

The general arrangement of the clutch is shown in Fig. 7. The clutch operating cylinder is placed alongside the engine with its center line parallel with the axis of the crankshaft.

The clutch cylinder is operated by air pressure and transmits its power through the crosshead, yoke, and fulcrum to the clutch.

The fulcrum bracket is of semi-steel and is carried on a foundation from the ship's floor. It is estimated that the load at the fulcrum is about 16,000 lb.

The clutch shifting ring is held in its position by the yoke pins and is lubricated for contact with the clutch shifting collar by means of a wick in the oil reservoir.

The engine-turning gear carried with the clutch consists of a worm wheel carried on the engine crankshaft flange and a hand-operated worm which is locked out of position when not in use. The diameter of the turning gear is $46^3/_8$ in., which is the extreme outside dimension of the assembled clutch.

Fig. 8 shows diagrammatically the arrangement of the clutch-shifting piping. Compressed air for the operation of the clutch

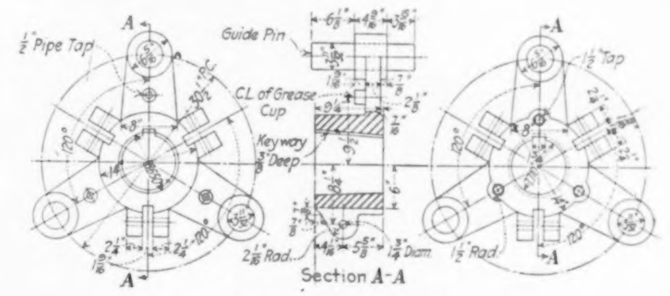

Fig. 4 Clutch Spider

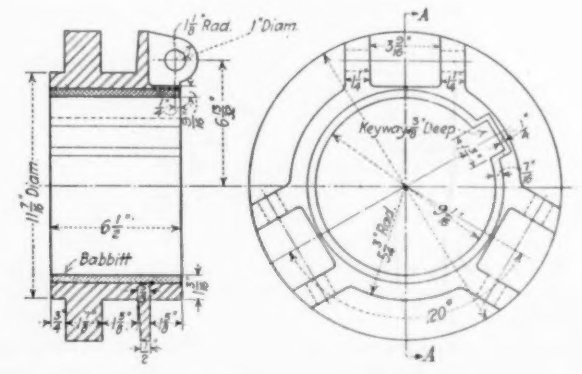

Fig. 5 Clutch Shifting Collar

is supplied from the main engine air-starting flasks at 160 atmospheres (metric measure) or about 2275 lb. per sq. in. It is noted that the air is drawn from the top of the flasks by means of an internal pipe to prevent water that may be collected in the flasks from passing through to the shifting cylinder and damaging it.

There is a pressure-regulating and relief valve in the pipe from the air-starting flasks, with stop valves on either side and at the flasks. The air passes through the clutch operating valve to the receivers and then to the clutch operating cylinder.

The clutch operating valve is so designed that its ports take care of the exhaust air from the opposite end of the clutch operating cylinder by a single movement of the handle, exhausting the air and entrained water to the ship's bilges.

When the clutch is thrown in, it is self-locking, as shown by position of links and toggles in lower half of Fig. 1. It is to be noted that this locking is accomplished by forcing center y slightly above centers x and z. Air may then be shut off from the clutch-shifting cylinder until it is necessary to throw the clutch out. The centrifugal force of the clutch will not throw it out. The advantage of the self-locking feature lies in the assurance that the holding power of the clutch does not depend on a constant pressure of air in the cylinder. Where air pressure would be used for holding the clutch in, there would be the possibility of leaks or fluctuations and the release of the clutch.

All air piping is $^1/_2$ in. in inside diameter and of seamless drawn steel with steel rings brazed to the ends for making up the joints. The pipes are secured in forged-steel fittings by means of a male nut placed on the pipe back of the brazed ring, the nuts then being screwed into the female joint of the fittings. Gaskets are either fiber or copper, placed within the fitting. All pipes and fittings tested to 3000 lb. hydrostatic pressure per sq. in.

For emergency there is a hand pump for the operation of the clutch.

Classification of German Diesel Engines and Clutches

The German Diesel oil engines were generally classified as follows:

- 10-cylinder, 4-cycle, 3000 i.hp., for large cruisers (submarine)
- 6-cylinder, 4-cycle, 1750 i.hp., for large operating submarines
- 6-cylinder, 4-cycle, 1200 i.hp., for large mine-laying and operating submarines
- 6-cylinder, 4-cycle, 550 i.hp., for mine-laying and coastal submarines.

Clutches for the 3000- and 1750-i.hp. engines were of the double-

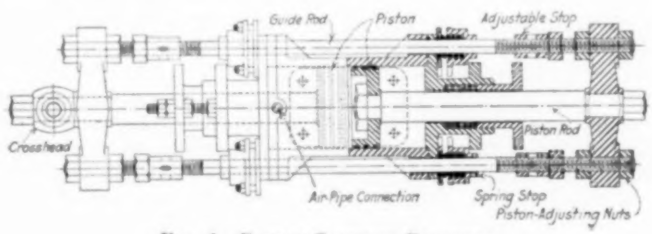

FIG. 6 CLUTCH SHIFTING CYLINDER

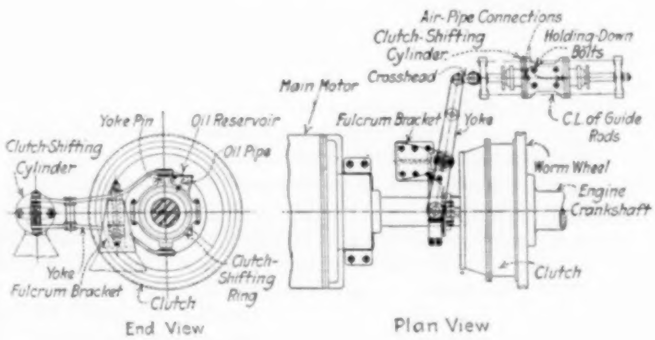

FIG. 7 GENERAL ARRANGEMENT OF CLUTCH

cone, friction, lubricated type, operated by compressed air or electric motor.

Clutches for the 1200- and 550-i.hp. engines were of the single-cone and disk, friction, dry type. The 1200-i.hp. engine clutches were operated by air and are described in detail in this paper. The 550-i.hp. engine clutches were operated by hand through worm and gear.

The details of design of the German Diesel-engine clutches were practically unknown in this country prior to the surrender of the German submarines, and considerable difficulty was experienced in dismantling them.

At the time of the arrival of the submarine freighter *Deutschland* at New London, when an appointed board of naval officers and civilians inspected her to establish her status, it is known that much interest was shown in her clutches, the construction of which was considered remarkable in so far as so small a diameter of clutch was capable of transmitting such large horsepower. No information of value was then obtainable, but later, when the surrendered German submarines arrived in this country, the clutches were one of the first mechanisms to be investigated.

It is believed that many improvements can be made in the design of American clutches by adopting some of the principles of the German clutches. The author has had some experience with American clutches and has found them to be of the tooth or positive-drive type, or of the friction-jack type. These clutches have been

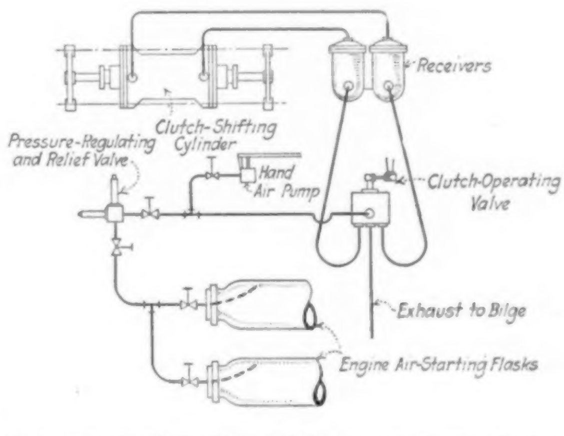

FIG. 8 DIAGRAM SHOWING ARRANGEMENT OF CLUTCH SHIFTING PIPING

the source of trouble and are constantly in need of repair. The tooth type does not allow the flexibility of shaft speeds required in maneuvering a ship that a friction type allows, and from experience it has been found that the friction types are not of sufficient strength and rigidity to transmit heavy loads. During the war one of our American submarines went below her depth by accident and it was necessary to insert emery paper in the clutch before the jaws would take hold and allow her motors to operate the shaft-driven bilge pump. Such delays may cause the loss of a ship and many lives.

The friction-jack types are usually thrown in by hand through a series of levers and therefore lack the holding power of an air-operated clutch.

With the friction-jack type of clutch there is nearly always a misalignment of shafts when the clutch is in, due to improper setting of the jack screws or adjusting screws. The screws are placed radially and must be adjusted from time to time to take up wear. It requires a skilled operator to properly adjust such a clutch.

In the German clutches the adjustment is fore and aft along the axis of the shaft, and with a proper alignment of shafts when machinery is first installed there is little possibility of misalignment when the clutch is in.

The author will be glad to give any further information desired by clutch manufacturers or engineers that will help to improve our American clutches.

During the past year there has been a marked drift toward larger units of Diesel engines. Only a few years ago the Diesel engine was confined to sizes under 500 hp. It was considered that 150 hp. per cylinder was the maximum to be obtained save at the sacrifice of reliability. This impression has been obliberated, and a number of manufacturers are building units of 300 to 400 hp. per cylinder. Two years ago an investigation by *Power* on the Diesel engine industry revealed that the average horsepower per engine was close to 300. In 1922 the average horsepower per engine sold was over 500 and the total horsepower approximately 70,000. This is evidence of the greater faith of the builder as well as the increased confidence of the power user in the internal-combustion engine. A marked activity has been shown in the development of solid-ignition, gas-injection, and other oil engines not using air compressors. These units have been of such sizes as to be competitors of the semi-Diesel rather than of the Diesel engine. It would seem that these types will find their logical place between the capacities that are naturally Diesel and those adaptable to the semi-Diesel. *Power*, January 2, 1923, p. 17.

Test Code on Instruments and Apparatus

Preliminary Draft of Chapters 1 and 2, Dealing Respectively with General Considerations and with the Accuracy of Measuring Instruments, and Being the First Installment of this A.S.M.E. Power Test Code to be Published

AS STATED a number of times before, the A.S.M.E. Committee on Power Test Codes is now engaged in the revision of the Society's Power Test Codes of 1915. Mr. Fred R. Low is Chairman of the Main Committee of twenty-five which guides the work of the nineteen Individual Committees. Below are reproduced the first two chapters of the Code on Instruments and Apparatus which is to cover the descriptions of instruments, apparatus, and processes common to the sixteen Power Test Codes. The individual Committee which is developing this Code is headed by Mr. C. F. Hirshfeld as Chairman and consists of Messrs. C. M. Allen, E. G. Bailey, L. J. Briggs, W. A. Carter, R. E. Dillon, S. B. Flagg, S. A. Moss, R. J. S. Pigott, G. B. Upton, E. B. Ricketts, J. J. Flather, F. M. Farmer and J. B. Grumbein.

The Committee and the Society will welcome suggestions for corrections or additions to these two chapters from those who are particularly interested in this part of the subject. These comments should be addressed to the Chairman of the Committee in care of The American Society of Mechanical Engineers.

CHAPTER I

General Considerations

1 The instruments and apparatus here considered are those used for measuring physical and chemical quantities in connection with tests of power equipment of various sorts, as required by other parts of the Code.

2 Measurement of a physical quantity never gives a result which is correct in an absolute sense. The numerical value determined always differs by some amount from the real value being measured, and the extent of the deviation depends upon the type of instrument and the method of its application. This fact imposes upon testing engineers the duty of studying measuring instruments and methods to such an extent that they can show in any given case that they have made all measurements to the degree of accuracy demanded by the purpose of the test.

3 The *accuracy* attainable in any given measurement is dependent upon four things:

(*a*) The method of applying the instrument
(*b*) The accuracy of the instrument itself
(*c*) The accuracy of the observer, and
(*d*) The characteristics of the quantity being measured.

4 As examples of the first effect the following may be cited:

(*a*) Even if a steam calorimeter were capable of indicating the quality of steam flowing through it without error of any sort and if the observer handled it perfectly, the value determined in any given case would not be the quality of steam in the pipe unless the sample flowing through the calorimeter correctly represented the steam flowing through the pipe from which the sample was drawn.

(*b*) If an absolutely accurate thermometer and a perfect observer could be obtained, such thermometer and observer would not necessarily give correct results in the measurement of, say, the temperature of superheated steam, unless the thermometer was properly immersed in the superheated steam or in a body having the same temperature as that steam and all necessary precautions were taken with regard to the thermal condition of the part of the thermometer not immersed.

(*c*) If a steam-engine indicator was a perfect instrument and perfect observers were obtainable, the card given by the indicator would still be in error if the pressure connections were not such as to apply instantaneously to the indicator piston the pressure acting on the engine piston, or if the motion of the indicator drum did not perfectly follow the motion of the engine piston.

5 In connection with the accuracy of the instrument, it should be noted that no instrument is accurate in an absolute sense. All instruments give only a more or less close approximation to the value of the quantity being measured. It is necessary to use that type which will measure to the required degree of accuracy and to know that it is in condition to do so. The accuracy of instruments will be considered at length in later paragraphs.

6 Accuracy of observers is not commonly given the attention it deserves. An observer should always be familiar with the theory and the mechanism of the instrument which he is using and should understand the degree of accuracy required on his part in its use. However, even after all such precautions have been taken there are still two forms of observational error to be guarded against. These are:

(*a*) Accidental errors, due to misreading the graduations, incorrect entry on the log of a value correctly read, failure to perform some necessary manipulation, etc.; and—

(*b*) Personal errors which are dependent upon the personal equation of the observer, such as carelessness in interpolation, tendency to consistently read high or low, inability to read rapidly, thus introducing a time lag, etc.

7 The characteristics of the measurement being made are of particular significance in engineering testing because conditions must usually be accepted as found. In the laboratory, on the other hand, conditions are ordinarily altered to facilitate measurement. Thus a quantity the magnitude of which is continuously changing is more difficult to measure at any moment than when it is not changing. The more rapid the variation the greater the difficulty, and, in general, the lower the accuracy attainable. Again, some quantities are measured so often under similar conditions that the methods have been developed in great detail, while others are measured so seldom or are ordinarily measured under such different conditions that little is known with respect to details, best methods, sources of error, etc.

8 At best, there will always be required in engineering tests certain measurements which are of such character that the degree of accuracy obtainable will be questionable. As examples of measurements which fall into this class at the present time the following may be cited:

(*a*) Determination of steam quality
(*b*) Determination of the temperature of superheated steam
(*c*) Determination of the calorific value of fuels
(*d*) Determination of combustible in boiler refuse, etc.

Since it is the purpose of the Code to provide methods upon which all can agree and upon the basis of which business can be done and guarantees can be made, rather than to produce a set of directions for scientific investigations, such cases are hereafter handled in a purely arbitrary manner. That is, attention is called to the known sources of error and to those precautions which can be taken to minimize their effects, and an arbitrary standard method of measurement is then prescribed. It is anticipated that as research develops, more accurate methods than these will be prescribed in subsequent revisions of the Code.

CHAPTER II

Accuracy of Measuring Instruments

1 Attention has been called to the fact that instrumental accuracy is purely a relative matter; there is no such thing as absolute accuracy in connection with the measurement of physical quantities. Accuracy in this connection has to do with two radically different considerations which may be designated as—

(*a*) The *intrinsic accuracy* of the instrument, and
(*b*) The *accuracy* resulting under the *conditions of use*.

The former is treated generally in the paragraphs which follow and more specifically in later sections dealing with individual instruments. The effects of conditions of use are considered in the sections dealing with the individual instruments.

2 *Accuracy* is used to designate the extent to which the indications of an instrument approach true values of the quantity being measured. The accuracy of an instrument can be determined by calibration, but it is important to note that this often determines the accuracy only for conditions of use similar to those maintained during calibration. For this reason it is often advisable to calibrate by methods which will subject the instrument to as nearly as possible the same conditions as those existing during use in a given test. It is also important to note that moving an instrument to or from the place of use or that dismantling during installation or removal may seriously affect its accuracy. For this reason calibration in place may be necessary under some conditions or with some types of equipment.

3 The methods used for expressing accuracy vary. One common method is to state that the instrument is accurate to within plus or minus a certain specific amount or a certain percentage at a certain point on the scale or between certain points on the scale. Thus a thermometer may be described as having an error not in excess of -0.5 deg. fahr. between 100 deg. fahr. and 300 deg. fahr. It is convenient to state the accuracy of instruments in this way when speaking in a general sense, but other methods are preferable for more specific applications. For such purposes the form of expression should indicate the correction to be added to the indication of the instrument to obtain the real value of the measured quantity. Thus, a thermometer might be said to have a correction of $+0.5$ deg., indicating that at this point of the scale it reads 0.5 too low. A correction of -0.5 would mean that it reads 0.5 too high.

4 For comparison of various instruments of similar character among themselves the specific inaccuracy at normal indications is a most useful value. The specific inaccuracy is the ratio of error to true value. The numerical value may be the same or different for various points within the range of the instrument. As an example of the meaning of this term, assume a pressure gage which reads 48 lb. when a pressure of 50 lb. is applied and 98 lb. when 100 lb. is applied. The specific inaccuracy at the lower point is $2/50 = 0.04$ or 4 per cent, and at the higher point $2/100 = 0.02$ or 2 per cent. If the inaccuracy is specified as a certain percentage of the full scale reading, then the maximum inaccuracy at any other reading is inversely proportional to that reading. This sort of thing is characteristic of nearly all measuring instruments which carry a uniform scale, that is, a scale divided uniformly from end to end. An error of given magnitude at a high scale reading represents a much smaller percentage error or specific inaccuracy than does an error of the same magnitude at a low scale reading. When instruments are so arranged that the length on the scale representing one unit of the measured quantity is long at the center of the scale and short at each end, the specific inaccuracy must obviously vary throughout the entire range in a more complicated way. Smaller inaccuracy in observation will be obtained with an instrument having two or more ranges than with one having only one range, as the deflection of the pointer or index to a point of maximum amplitude will reduce to a minimum errors due to parallax in observation or to any other cause.

5 *Inaccuracy* of an instrument in the sense in which it is here used is caused by features or factors in the instrument itself and has nothing to do with the effect of external conditions other than the value of the quantity being measured. Inaccuracy may be due to great number of different causes acting singly or in combination. The principal causes are:

(*a*) *Imperfect material*, as variation in bore of a glass tube, variation in the molecular structure of the material of springs, etc.

(*b*) *Unavoidable physical phenomena* such as capillary attraction, imperfect refraction of light, friction of rest and motion, etc.

(*c*) *Imperfect construction* which *cannot be improved*, such as lost motion in gear trains resulting from necessary clearance in bearings or between teeth, changing lengths of levers due to necessary clearance in bearings or to the finite width of knife edges or points, etc.

(*d*) *Imperfect construction* which *can be improved*, such as excessive friction, excessive lost motion, incorrectly cut cams, etc.

(*e*) Unavoidable or partly unavoidable *properties of materials* such as the aging of glass, the gradual yielding of materials to physical stress, the aging of permanent magnets, etc.

6 Inaccuracies arising from such causes as are included under Par. 5 (*a*) are ordinarily of irregular character, causing more or less erratic errors throughout the range of the instrument. They are usually permanent in character or change but slowly and are easily taken into account during calibration.

7 Inaccuracies arising from such causes as are included under Par. 5 (*b*), (*c*) and (*d*) are more or less regular in character and are responsible for what has been called the hysteresis loop of an instrument. This is the loop which is obtained by plotting instrument readings against true values, using the instrument first in an ascending sense and then in a descending sense. A similar but different loop can be obtained by plotting errors against true values or corrections against indicated values. This is considered later in greater detail.

8 Inaccuracies arising from causes included under Par. 5 (*e*) are responsible for what is known as aging or drift in instruments. Within certain limits such aging or drift is not detrimental, although it does necessitate frequent calibration to guard against its effects. In certain cases it must be taken into account if erratic results are to be avoided. Thus a glass thermometer which is to be subjected to high temperatures must be artificially aged before calibration and use.

9 *Sluggishness* is a term used to indicate the amount of displacing or actuating effect required to cause motion of the indicating part of an instrument. It is determined by noting the smallest alteration in the quantity being measured that will produce a perceptible change in the indication of the instrument. Thus a thermometer which at 400 deg. fahr. requires a change of ± 5 deg. fahr. to cause it to move up or down will have a sluggishness index of $5/400 = 0.0125$. This factor is most useful when expressed as a percentage, giving 1.25 per cent in the example cited. In comparing the sluggishness-index factors of instruments calibrated in different units it is necessary to refer them to the same base. Thus a thermometer calibrated in degrees centigrade would have the same factor as the one just considered if it required a change of ± 2.78 deg. cent. ($= 5/1.8$) to cause it to move up or down, since the temperature of 400 deg. fahr. corresponds with 204.4 deg. cent. or 222.2 deg. cent. above 0 deg. fahr. (-17.8 deg. cent.) which is the base of the fahrenheit scale. The sluggishness factor in this case would be $2.78/222.2 = 0.0125$ or 1.25 per cent. The value of the factor may and probably will be different for all points within the range of the instrument. "Passiveness" is often used as synonymous with sluggishness.

10 Sluggishness is often thought of as the opposite of sensitiveness or sensibility. Thus it is common to speak of a thermometer as "sensitive to 0.1 deg. fahr." meaning that it will respond to changes of that magnitude. This is technically an incorrect usage. Sensitivity is defined as the rate of displacement of the indicating element of the instrument per unit change of the measured quantity. That one of a group of comparable instruments which gives the greatest displacement for a given change in the measured quantity is the most sensitive.

11 *Sensitivity* of instruments is a most important property, as with other things equal it is the thing which determines the degree of refinement possible in their use. Unfortunately instruments are not always designed and constructed in such a way that the total available sensitivity is useful. Thus an instrument may be so constructed that it is possible to read on its indicator the movement corresponding to a variation of one unit in the value of the quantity being measured, but it may be so built that it will indicate values varying by as much as five units in successive trials with no change in the actual value being measured. The sensitivity of such an instrument is partly useless and is really misleading. A correctly designed instrument should have such sensitivity that the minimum readable movement of its indicator is not less than the amount by which the instrument may be expected to vary in successive indications of the same value of the measured quantity. To be exact these requirements should be stated in terms of the mean for both indication and variation as both may vary from point to point throughout the range of the instrument.

12 Sensitivity in excess of requirements, as set by the accuracy

desired in a given set of measurements, is generally a detriment because it involves delicacy, slowness of operation, or some other undesirable quality.

13 *Variance* is a term used to represent the amount by which the readings of an instrument vary in successive indications of the same value of the measured quantity. Variance is due principally to—

- (*a*) *Lost motion*, which also affects the sluggishness or passiveness;
- (*b*) *Friction*, which also affects the sluggishness or passiveness;
- (*c*) Changes due to the stress-strain relation of springs *in the force-resisting or restoring element* of the instrument;
- (*d*) *Changes in the distribution of parts*, as variation of position of pins in bearings or variation in the amount of liquid retained on wetted surfaces
- (*e*) The *immediately preceding history* with respect to extent and speed of displacement; and—
- (*f*) A great number of other more or less obscure phenomena.

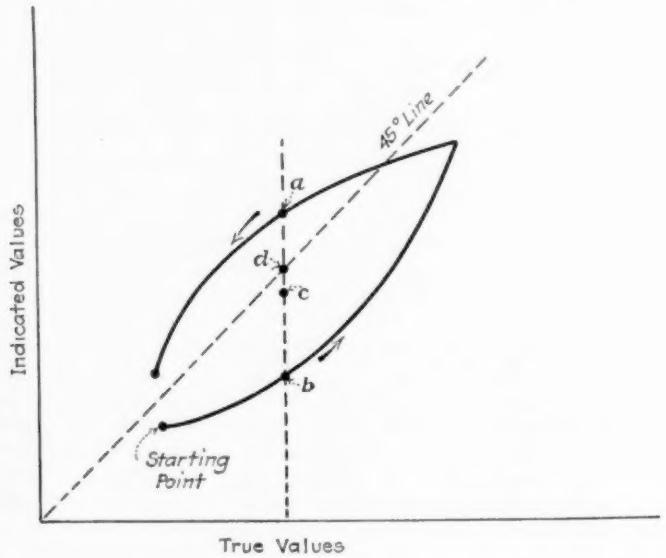

Fig. 1

14 Variance due to lost motion is commonly associated with the direction of motion of the parts in the instrument and it could be almost entirely eliminated if a given value could always be approached from the same side. Under conditions of use this is commonly impossible, although it can be done during calibration if desired.

15 Variance due to friction can be partly eliminated by tapping the instrument just before taking a reading. This is common practice in the use of instruments containing mechanical movements and is recommended if not carried to excess.

16 Variance due to the other causes enumerated above is not under control except in so far as it can be kept at a minimum value by proper maintenance and proper handling.

17 Instruments which are used in tests made according to the A.S.M.E. Power Test Codes must be tested to such an extent as may be necessary to determine their condition and their accuracy. Such testing is commonly known as *calibration of the instrument.*

18 The exact procedure to be followed in calibrating instruments varies not only with the instrument but also with the use to which it is to be put, the accuracy desired and other considerations. In general the calibration should include determination of the following throughout the range which is to be used:

- (*a*) Sluggishness and its relation to the sensitivity of the instrument
- (*b*) Variance and its relation to the sensitivity of the instrument
- (*c*) Accuracy, or error of indication at different points within the useful range.

19 The principal methods available for the calibration of instruments and apparatus of different sorts are treated in detail in later paragraphs when the individual instruments are considered. Certain general features more or less applicable to all instruments are, however, discussed in the following paragraphs.

20 Calibration for the determination of error of indication may be conducted with one of two different objects in view: It may be desired to determine the errors of indication with respect to absolute accuracy; or it may be desired to determine only the extent to which the instrument is constant within itself with no reference to absolute values. As an example of the latter case, consider a speed-measuring device used in connection with a test to determine governor regulation with change of load. Under such conditions it is essential that the instrument shall give properly comparable values, but a correct indication of any one speed is not essential. In most cases, however, calibration on an absolute basis is required.

21 Most *calibrations* for error of indication are made by direct *comparison* of the readings of the instrument under calibration with the readings of some other instrument which is used as a standard, or by subjecting the instrument to the action of effects of known magnitude. Thus a thermometer may be calibrated by comparing its indications with those of another thermometer previously calibrated, with the two immersed in a liquid under such conditions that they should indicate the same temperature. Or, a thermometer may be calibrated by immersing it in pure materials undergoing changes of physical state known to occur at certain definite temperatures.

22 When it can be done, it is best to calibrate by some method which will make it possible to determine the errors of the instrument at successive points on its scale while the quantity being measured is varied by steps in an ascending and in a descending sense, successively.

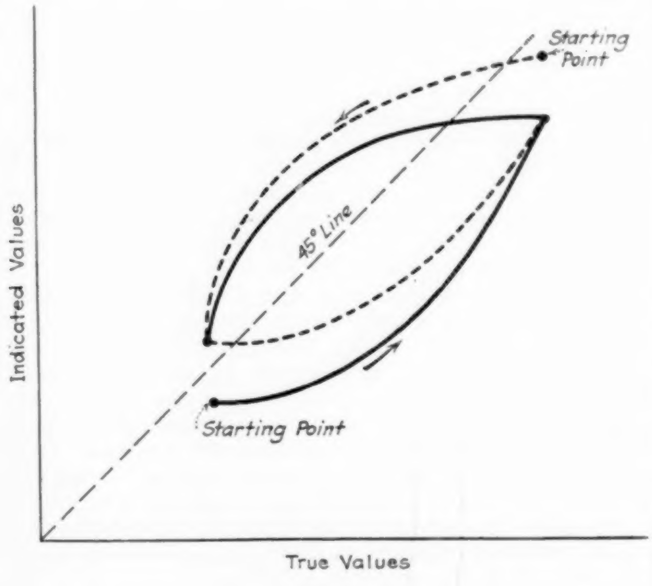

Fig. 2

23 When indications obtained in this way are plotted against true values a *hysteresis loop* will generally result if the work is done with sufficient care and skill. Such a loop is shown in Fig. 1 for a case in which ascending values are determined first and in which the instrument was reading a lower value than that at the starting point before calibration started. In such a case the lower end of the loop will not commonly close. If the procedure is reversed so that calibration starts at the higher values the resultant curve will be reversed in shape and the upper end will not close in most cases.

24 An ideally perfect instrument absolutely accurate throughout the range under test will give results plotting on the 45-deg. line indicated in Fig. 1 if true values and indicated values are plotted to the same scale.

25 In Fig. 2 are shown two hysteresis loops which might be obtained when calibrating the same instrument. The loop shown in full lines would be obtained with ascending values determined

first and that shown in dotted lines with descending values determined first. Obviously there is no simple correction for such an instrument. The departure from the true value varies with the past history of its moving parts or elements. It is also important to note that successive trials may not give values which will plot on the same hysteresis loops, particularly if the steps used in successive trials are of radically different magnitudes, or the rates of variation are radically different or the amount of vibration to which the instrument is subjected differs.

26 For convenience in engineering tests it is customary to draw some single line or curve as a sort of average of the hysteresis loop and to use this as the "calibration curve" of the instrument. This line or curve is usually drawn by plotting a series of points, each one of which is the mean of figures obtained for corresponding true values when ascending and descending. This procedure is shown in Fig. 3 for the case in which only one hysteresis loop is determined. It is obvious that the exact shape and location of such a curve will vary with the way in which the loop is obtained. It follows that the "calibration curve" is not a true representation of the behavior of the instrument during its calibration, and, further, that it is not an exact means for correction of indications obtained under the constantly varying conditions of use. The approximate nature of the calibration curve should be kept in mind in all considerations dealing with the accuracy of observations and hence with the accuracy of test results.

27 It is obvious that, with other things equal, the most accurate results are to be expected with instruments giving the narrowest

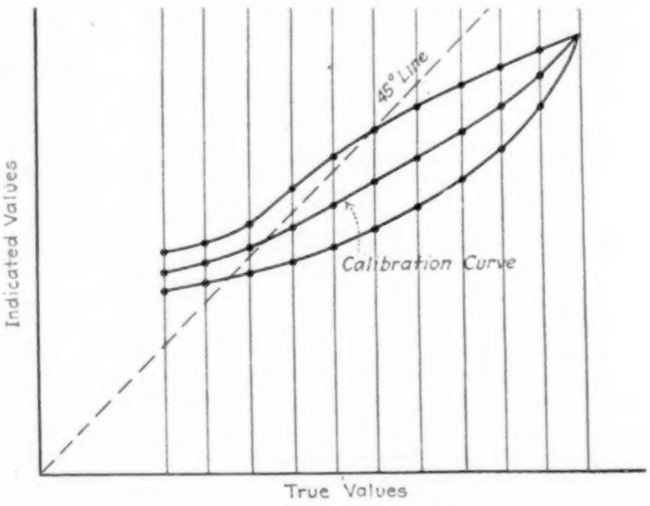

Fig. 3

hysteresis loops, and such instruments should be used when obtainable and when their use is justified by the degree of accuracy desired in the test.

28 The calibration curve as shown in Fig. 3 is not in the most convenient form for use under ordinary conditions. It is better to put it in the form shown in Fig. 4. Drawn in this way it is easily used and there is little possibility for error in its use. It has the further advantage that for ordinary conditions the entire plot occupies very little space, so that a number of curves can be drawn on one sheet without danger of causing confusion. Such an arrangement is shown in Fig. 5.

29 It will be observed that the calibration curve obtained in the way indicated shows only the average amounts by which the indications of the instrument may be expected to vary from the true values of the measured quantity. Such a curve tells nothing about sluggishness, sensitivity, or variance. These characteristics are, however, of importance in most cases and should be investigated during the calibration for the purpose of determining the adaptability of the instrument to the use intended, its condition, and the accuracy which can be obtained in its use.

30 *Sluggishness* is best investigated by determining the amount of displacing effect required to start useful motion of the indicating element after the instrument has been at rest. Such determinations should be made throughout the range which it is intended to use. In most cases such tests can be made conveniently while obtaining data for the hysteresis loop, so that a separate calibration for determining sluggishness is not commonly necessary. The observations made in determining sluggishness should be recorded with the other calibration data. They are best converted to a sluggishness index for successive points in the scale and tabulated as such.

31 The *sluggishness index* is not an exact numerical value in an absolute sense. The amount of vibration to which an instrument is subjected during calibration or use has a marked influence on its performance in this respect. Vibration has a tendency to decrease

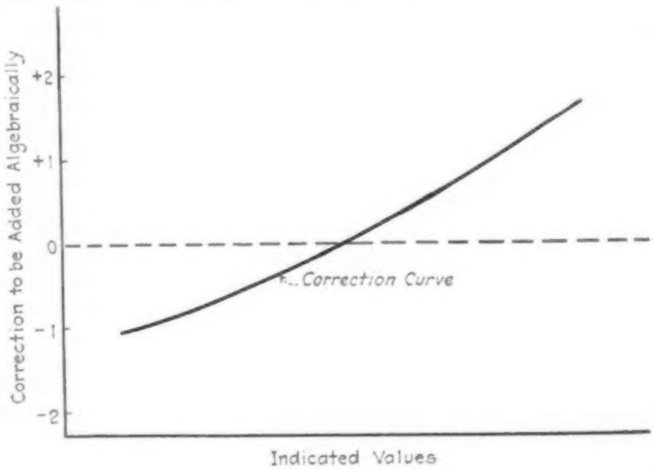

Fig. 4

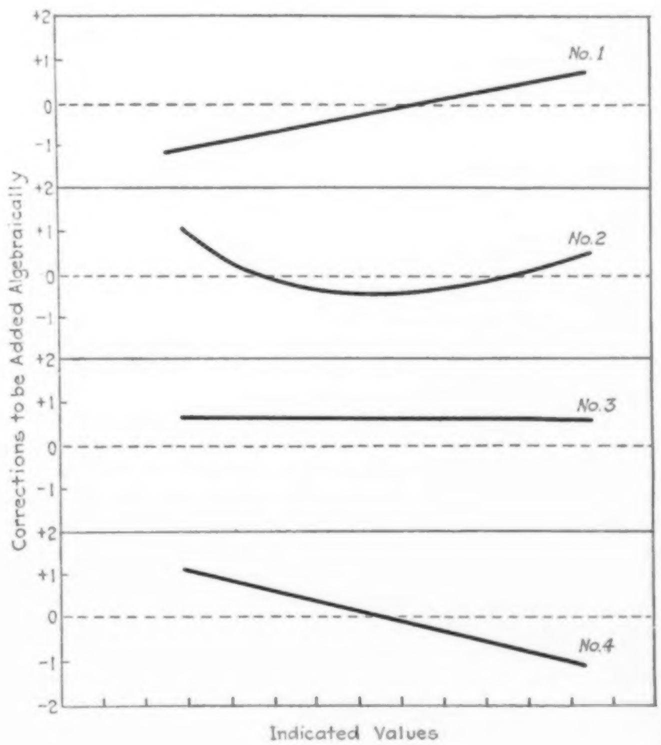

Fig. 5

sluggishness in all instruments containing kinematic trains of any sort, pivots, bearings, and the like. For this reason it is customary to tap such instruments to induce vibration before reading. This should be taken into account during calibration, both in determining the error of the instrument and in determining the sluggishness factor. If an instrument is going to be used under such conditions that it will be subjected to vibration or can be tapped before reading, it should be calibrated under analogous conditions. Again, the rate of variation of the measured quantity has a marked effect on sluggishness. The sluggishness factor determined when the variation of the measured quantity is very slow and very steady will be, ordinarily, quite different from that determined when the variation

is rapid but steady and, further, quite different from that determined with rapid variations of opposite sign. All these factors should be taken into account when calibrating, an effort being made to approximate as nearly as possible the conditions to which the instrument will be subjected when in use.

32 *Variance* can be determined directly from a hysteresis loop or, better, from a number of hysteresis loops obtained by starting at different points and by subjecting the instrument to variations of different directions, intensities, etc. The points marked *a* and *b* in Fig. 1 show two radically different indications which might be obtained for the same value of the measured quantity. It is quite common to assume that a point *c* midway between points *a* and *b* indicates the reading that would be obtained were there no variance. This is consistent with Par. 26. With this assumption the distance between *a* and *c* is a measure of the variance of the instrument at this particular point in the scale. There is obviously no certainty that the instrument will ever read closer to *c* than is indicated by the points *a* and *b*, respectively, and the possibility of variance of such extent must always be assumed in testing. Naturally, the value of variance will vary from point to point, so that it should be determined throughout the range that is to be used. When variance is determined in this way it should be tabulated or plotted and the record preserved with other calibration records of the instrument.

33 *Variance*, like sluggishness, varies in value with the conditions existing during test or use. The character of the variation of the value of the quantity being measured the amount of vibration and other factors all have their influence. As an example, consider an instrument subjected to a sudden increase of actuating force or effect of sufficient magnitude to cause its indicating parts to move so rapidly that inertia carries them beyond the position they would normally assume, for the final value of the quantity being measured. It may be that the amount of overtravel is so great that the unbalanced forces resulting are more than sufficient to overcome sluggishness without any vibration of the instrument. It may be that the overtravel is not so great and that sluggishness prevents restoration to the proper position even with vibration. It may be that the lost motion in the mechanical train is so great that even though the essential moving part of the mechanism comes back to a correct position the indicator or pointer remains in the position of overtravel. There are so many possibilities that it is practically impossible to correct for variance in the actual use of many instruments. The best that can be done is to recognize the fact that the calibration curve of the instrument is in reality not a line but a band of finite width (or height) and to interpret its readings and test results depending on those readings in the light of such knowledge.

34 *Sensitivity* is a property determined by the design of the instrument; it does not change with the condition of the instrument. In case of mechanically operated instruments if a certain moving element has been chosen and a certain multiplying device has been designed, the pointer or other indicator must theoretically move a certain distance for a given change in the value of the quantity being measured. In the case of a device like a mercury thermometer, if a given amount of mercury is subjected to a given change in temperature a certain volume change must occur and, in a capillary tube of a given bore, this must cause a movement of a given amount. The numerical value of the sensitivity is thus a characteristic determined by the design of the instrument. It is significant primarily in making a choice of instruments. These must have such sensitivity that the values of the quantities being measured can be read upon the instrument to at least as small a subdivision as is required by the conditions of the test.

35 The value of the sensitivity can be determined readily from the results of the calibration by noting the amount of displacement occurring for given variations in the values of the quantity being measured. It then becomes possible to state the displacement per unit of value and the minimum readable variation which follows from the design of the instrument.

36 It is always necessary to compare the values of sensitivity thus determined with the value determined for sluggishness and variance. Sensitivity within the limits set by sluggishness and variance is useless, and an attempt to use it results only in wasted effort and, possibly, mistaken accuracy.

LUMBER DRY KILNS

(*Continued from page 112*)

A few standards of construction, design, and operating methods are given below for the purpose of assisting in general layouts and development sketches.

Kilns may be *end-pile* or *cross-pile*, i.e., boards may be parallel or at right angles to the rails (Fig. 4), depending largely on the factory and yard arrangements. End-pile gives slightly better internal circulation, although this is rarely determinative.

A kiln car of lumber is usually 6 ft. wide, 9 ft. high (above rails including trucks), and from 12 to 16 ft. long (rarely longer). Higher loads not only upset easily but greater heights introduce uneven drying, or wet pockets, because of unavoidable temperature differences between the bottom and top of an enclosed space. The writer knows of a battery of kilns 20 ft. high in which the top half of the load always dried first and there was a temptation to discharge the kiln before the bottom half dried.

Kilns may be of the *progressive* (open at both ends) or *single charge* (usually open one end) type as shown in Fig. 5. The former

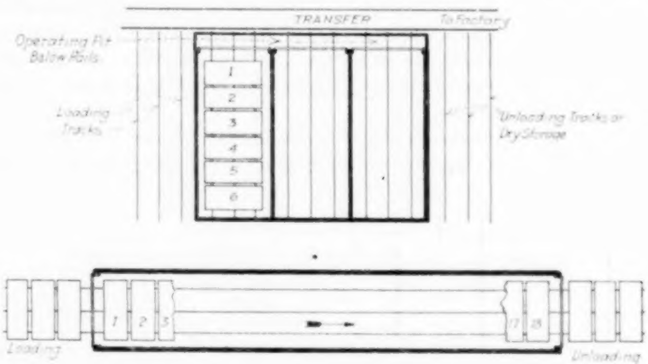

FIG. 5 PROGRESSIVE KILN (BELOW) AND SINGLE CHARGE (ABOVE)
(Each has equal holding capacity and output, together with minimum necessary trackage for loading and unloading.)

involves a daily opening to remove and enter lumber, a daily moving forward of every car, with a consequent loss of from 5 to 10 per cent of the drying period from the opening of kiln, during the time required to cool sufficiently to move the cars, to the time when temperature and humidity are restored to operative conditions. The progressive kiln, to be efficient, requires that the charging end be maintained at a low temperature and a high humidity and the discharging end at a high temperature and low humidity with graduated intermediated conditions. Air circulation, chiefly transverse, prevents dependable graduated conditions and the operator copes with irregular conditions. The subdivision of the heat and humidity sources may be of slight help, but in any case the progressive kiln is more complicated than the single-charge type.

The latter type (sometimes called box, compartment, pocket, or cell) requires the operator to vary his entire kiln regulation from day to day according to a predetermined schedule. The room is small enough so that control of uniform conditions throughout is simple. Various kinds and thicknesses of lumber can be separated in different rooms and given individual treatment. Valves and dampers can be located in an accessible outside pit. When built in batteries with adequate transfer and yard trackage the single-charge type will produce as great a turnover from the investment as the progressive, and dry the lumber better.

Hollow tile, with an exterior brick veneer, makes the best walls, and a tile and reinforced-concrete roof with insulated wooden doors will complete an economical heat-retaining building. Solid concrete is not only a poor insulator but adsorbs too much moisture and checks badly under high temperatures. Frame buildings, under the varying conditions of temperature and humidity, rot and rack rapidly.

Scientific kiln drying is only at its beginning and wood users have but commenced to visualize the efficiency and economy that is possible. Engineers can do much toward the solution of the problem by giving it the same degree of study and research that has resulted in substantial progress in other fields.

SURVEY OF ENGINEERING PROGRESS

A Review of Attainment in Mechanical Engineering and Related Fields

AERONAUTICS (See also Internal-Combustion Engineering)

British Air Ministry and Rateau Scheme for High Flying. It is stated that the Directorate of Research of the Air Ministry (British) has rejected the Rateau scheme for flight at high altitudes for the following reasons quoted from the letter of the Directorate.

"Considering the conditions under which both turbine and blower have to work, it is not considered possible to obtain overall efficiencies of the order of 30 per cent. A more likely value would be 20 per cent, with 25 per cent as a practical limit. Taking 22.5 per cent as a value for an average plant, back pressures of the order of 3 lb. per sq. in. will have to be allowed in order to restore ground-level inlet pressures at 15,000 ft. The utility of the device will then depend largely on the effect this back pressure has on the running of an engine. In this connection both loss in power and possible overheating will have to be considered. In any case, the strictly limited range of 'full compensation' shows that the scheme proposed by Professor Rateau is not a complete solution of the problem, and that at greater altitudes, especially, other schemes will have to be considered." (*The Practical Engineer*, vol. 66, no. 1867, Dec. 7, 1922, pp. 353–354, *g*.—The mathematical foundations for the high-altitude flight scheme of Professor Rateau, Mem. Am.Soc.M.E., were presented in his paper before the Society and published in Mechanical Engineering, May, 1920.)

Paragon Adjustable and Reversible Propeller. Description of a propeller recently tested at Bolling Field. For this purpose it was installed on an automobile truck trailer equipped with a 150-hp. Wright-Hispano engine.

The two wooden or steel blades are fastened into steel sleeves, which, in turn, are held in a steel hub, the centrifugal force being taken on ball thrusts and torsional and axial forces on plain bearings. In the method of fixing the wooden blades into the steel sleeves as illustrated in the original article, the butt end of each blade is tapered outwardly at a small angle and the surrounding collar is split so that it may be first sprung over the butt and then compressed into the taper.

The pitch-changing mechanism is operated through the application of a braking force to either one of a pair of small brake drums surrounding the engine crankshaft and normally rotating with it. The brake drums are connected through a gear train to the individual blades of the propeller.

In the tests the engine was turning at 1500 r.p.m. No data are given as to the strength of the blade where it is held by the split collar and as to its ability to resist the centrifugal stresses, which is the main problem in the design of a variable-pitch propeller. (*Aerial Age*, vol. 15, no. 21, Dec., 1922, pp. 587–589, *d*)

Four-Engine All-Metal Airplane Built by the Schneider Co. Description of a airplane of the Henri-Paul type built by the great Schneider Works, the largest steel plant in France. There are several reasons why this airplane becomes of considerable interest.

The machine is of very considerable dimensions, weighing with full flying kit 10,020 kg. (22,050 lb.). It is equipped with four Lorraine-Diétrich motors of 370 to 400 hp. each.

The wings are made with chrome-nickel steel frame members with stiffening ribs of a light alloy. The fuselage is made up of high-resistance light alloy and steel tubing.

The original article gives considerable attention to the aerodynamic properties of the machine and some data on design. It is also stated that the Schneider works have under construction two four-engine airplanes which are each equipped with a 75-mm. gun. (*Le Génie Civil*, vol. 81, no. 23, Dec. 2, 1922, pp. 505–510, 18 figs., *dA*)

ELECTRICAL ENGINEERING (See Power-Plant Engineering)

ENGINEERING MATERIALS

Lead-Coated Pipe in the A. T. & S. F. Railway Enginehouses. Considerable trouble has been experienced from rapid deterioration of water pipe in enginehouses on the A. T. & S. F. Railway, and in an effort to remedy the situation a process of lead plating of pipe has been developed.

This is done in the following way: The material to be plated is first cleaned by immersion in a lye vat which serves to remove the grease and oil. The surfaces are next washed with a jet of water and then placed in another vat filled with a 17 per cent solution of sulphuric acid, which is followed by scouring produced by adding about 15 lb. of Chile saltpeter for each 600 sq. ft. of treated surface. The saltpeter is added after the steel has been placed in the sulphuric acid solution and its addition is followed by a violent eruptive action on all metal surfaces within the vat, which, by the way, is constructed of redwood timber. After this action has subsided the steel is removed from the cleaning solution and again washed with a jet of water preparatory to the plating operation. Lead plating is effected by dipping the steel in a vat filled with lead acetate solution ranging from 20 to 50 per cent for a period of about 20 min. About 0.1 oz. of lead is deposited per square foot of steel and the weight of this lead plating can then be built up electrochemically if desired.

It is stated that the resultant material retains all the physical strength of the steel superstructure and is as impervious to corrosive action as pure lead.

The article describes also the new method of making pipe joints by means of forged steel flanges welded to the tube ends. (*Railway Review*, vol. 71, no. 24, Dec. 9, 1922, pp. 818–820, 3 figs., *d*)

Carbonized Clay—A New Refractory Material, Walter Smith. General discussion of the carbonization of clays with special reference to the osmose process and material produced thereby.

In general carbonization of clays is based on the fact that clay reaches maximum porosity in the earlier period of firing at a point when the clay is in the biscuit state. The material is then extremely absorbent. This opportunity is seized for charging the clay with volatilized hydrocarbon. As the heating is continued the clay powerfully contracts and entraps the particles of atomic carbon which are then compressed from the easy position adopted by themselves by volatilization to a great density. The product thus made is carbonized clay.

Black carbonized clay consists of clay grains reinforced with countless infinitesimal particles of carbon diffused throughout the pores. This material is remarkably dense, and so long as it remains in a reducing atmosphere is capable of withstanding the destructive action of heat beyond the highest commercially workable temperature and is unaffected by acids.

White carbonized clay results from the action of an oxidizing flame on black carbonized clay. When such a flame plays on the black material myriads of particles of enclosed carbon are gradually consumed, a corresponding number of minute pores being created.

Thus, instead of having the porosity which occurred naturally in the biscuit state of the clay (uncontracted), there exists an artificial (and almost equivalent) porosity in the fully contracted material, and, since the contractive force of the clay has been expended, the artificially created pores remain constant. In other words, when further heating is applied to the mass, the limit of contraction having been reached, difficulty is encountered by the heat in bringing about a state of fusion; the fusing point is therefore transferred several hundred degrees higher than that of the same clay in its natural state.

The black state provides a material adapted for heat resistance in reducing atmospheres, for the manufacture of acid-resisting ware, and for abrasive powders. In the white condition the clay should be used for the manufacture of refractory materials to be employed in the open flame, and in furnaces where air is freely admitted.

It is stated that the material is very little affected by hot acid fumes, both hydrochloric and sulphuric acid. The hardness of carbonized clay is just short of carborundum; indeed, it might replace the softer grades of carborundum and the material is amorphous instead of crystalline. Its heat conductivity is twice that of standard firebrick. A specimen was shown which was intact after resisting a temperature of 1730 deg. cent.

A remarkable phenomenon which occurs as a result of carbonizing clay is the fact that by carbonization iron oxide may be liberated from clay as a by-product. Alkalis are similarly liberated. (*Journal of the West of Scotland Iron and Steel Institute*, vol. 30, no. 1, Oct., 1922, pp. 8–13, *dA*)

Marine Borers Attack Piling Along Atlantic Coast. In the report of the San Francisco Bay Marine Piling Committee (compare Mechanical Engineering, Dec., 1922, p. 834) it was mentioned that the marine borer had appeared on the Atlantic coast. It is stated now that the National Research Council has undertaken an investigation to determine methods of protection. The first appearance on the East coast was established in Barnegat Bay, New Jersey, in 1921. In order to secure definite information regarding the distribution of the various species and their rate of growth, a test board was designed carrying 24 blocks, one of which was to be removed on the first and sixteenth day of the month and sent to Harvard University or the University of California for proper investigation. The blocks are 2 in. by 4 in. by 5 in. in size, of whitering sappy yellow pine, surfaced on four sides. The first boards were placed in June, 1922, and altogether 235 of these boards have been put in place between the eastern boundary of Maine and Kodiak, Alaska. As a result it was found that the various types of borers are present practically along the entire coast.

In an effort to develop a system of protection, test blocks impregnated with various chemicals, in particular fractions of creosote, are being tested by immersion in Gulf and Pacific harbors, several tropical woods being also included in the test. A report on this work has not been published as yet. Reinforced concrete as a substitute for timber has not a record of useful success; in fact, the Committee has found very many more cases of failure than of success, and in most cases the causes of failure are very obscure.

This investigation is being conducted by the Committee on Marine Piling Investigations of the National Research Council, New York City. (*Railway Age*, vol. 73, no. 24, Dec. 9, 1922, pp. 1083–1085, 5 figs., *g*)

FOUNDRY

Casting of Aluminum Bronze, Chas. Vickers. Aluminum bronze requires some special methods of casting as compared with the practice of a brass foundry. It drosses and shrinks excessively and is also subject to gas cavities and internal shrinkage holes. Furthermore, machining is apt to disclose little nests of a white non-metallic substance (alumina). In the case of castings massive in size the alloy changes as it cools slowly in sand molds and suffers thereby a drop in tensile strength and particularly in elongation. (This change, known as "self-annealing," has to be counteracted either by some device for the quick chilling of the casting or by the use of special compositions of aluminum bronze immune to the effect of slow cooling.)

The article describes in detail how to counteract these undesirable tendencies in castings, in particular the method of gating used. Among other things it is recommended to use horn gates with the large end toward the casting and heavy risers to feed the shrinkage. Spiraling of the down gate is also suggested, and it is stated that a spiral runner of small cross-section is an important means of securing dross-free castings of aluminum bronze, because it eliminates vertical sprues of large size down which it is impossible to drop the liquid bronze and not create an excessive amount of dross.

The term "aluminum bronze" can be taken as meaning any alloy of copper and aluminum, with or without additions of other elements which have a base of copper. An alloy commonly used for small castings is copper, 89 per cent; aluminum, 10 per cent; and iron, 1 per cent. This has some advantages over the alloy of 90 per cent copper and 10 per cent aluminum. Where great strength is required, as in the case of large valve stems, shafting, etc., an alloy of copper, 86 per cent; aluminum, 10 per cent; iron, 4 per cent, is extensively used. Since this alloy was devised by the author in 1916 it has been extensively used under a variety of names, a proof of its remarkable qualities. In 1918, patent No. 1,264,459 was granted from the application filed Sept. 27, 1916. Its physical properties will vary according to the skill with which it is made. In tensile strength the cast metal should reach considerably over 80,000 lb. per sq. in. and its elongation should not be under 17 per cent, sometimes running as high as 30 per cent. (*The Foundry*, vol. 50, no. 23, Dec. 1, 1922, pp. 958–960, 4 figs., *d*.—In this connection it may be of interest to note that the first comprehensive account of the properties of aluminum bronze published in this country was printed in the Transactions of The American Society of Mechanical Engineers, vol. 15, p. 631, discussion by Mr. Leonard Waldo, Mem. Am.Soc.M.E.)

FUELS AND FIRING

Factors Affecting the Use of Air in Oil Burning, with Comparison of Cost, W. C. Buell, Jr. Discussion of the influence of air pressure in burning liquid fuels. The author classifies all oil-burning systems offered for industrial heating practice into three general types on a basis of air pressure at the source of supply from which they operate, these three classes being: (1) Burners employing so-called "volume" air which is moved by high-speed fan blowers and delivered to the burners at pressures of from 4 to 10 or 12 oz.; (2) burners operating on so-called "positive" air which is delivered to the burners by displacement or turbine-type machines and occasionally by high-speed centrifugal fans, at pressures of from 12 oz. to 2 lb.; (3) burners utilizing "high-pressure" air which is delivered to the burners from compressor lines in which pressures ordinarily run from 50 to 60 or 100 lb. gage.

The author discusses the factors affecting the use of air in oil burning under the above classifications, but with reference to the equipment included under class 1, it should be understood that burners operating on the low pressures of this class cannot depend on the air to atomize the oil but must of necessity use a mechanical system in which the oil fuel is atomized by the static oil pressure forcing the oil fuel through very small orifices in the form of a fine spray which the low-velocity air can easily pick up. The use of this system is therefore limited to oil which is very fluid and very clean. This confines it to a comparatively small field, as in this day and age only residual and heavy fuel oils should be considered as fuels.

The investigation of the author covers the subject of air velocity, quantity of air required for burning oil, inductive properties of primary air as applied to oil-burning practice, comparison of blowers, and, finally, comparative costs of air.

The author comes to the conclusion that consideration of the subject-matter and figures of this paper clearly show that the proper method of burning oil is with air pressure under classification 2, or under some conditions with 15-lb. air under classification 3.

Further, all fuels are high priced whether they are coal, natural gas, manufactured gas or the liquid hydrocarbons, and in the years to come the tendency will probably be for fuel prices to continue on an upward trend; therefore, to cut down fuel costs to the lowest possible figure it is necessary that all of the economic factors entering into combustion reactions be given the most careful consideration and sound engineering principles applied to the installation, application, and operation of all fuel-burning equipment.

The conclusion drawn throughout the paper can be applied with only slight modifications to the combustion of pulverized fuel. Naturally the requirements for atomizing as with oil are present only to a small degree, but undoubtedly the practice is similar with both types of fuel.

No attempt has been made to cover oil-burning equipment as used in the firing of steam boilers and open-hearth, glass, or other regenerative or recuperative furnaces. Conditions of combustion

set up with the latter type of equipment vary so radically from those found in industrial-furnace application that the results given in this paper can in no way be applied. (*Proceedings of the Engineers' Society of Western Pennsylvania*, vol. 38, no. 6, July, 1922, pp. 201–221, 11 figs., *epA*)

HANDLING AND CONVEYING (See Special Machinery)

HEATING (See Thermodynamics)

HEATING AND VENTILATION (See Power-Plant Engineering)

INTERNAL-COMBUSTION ENGINEERING (See also Motor-Car Engineering)

RESEARCH ON SMALL DRILLED ORIFICES, W. S. von Bernuth. Data of tests carried out at Purdue University with special reference to the extremely small holes needed in carburetors and vaporizing devices, including a description of the special apparatus designed for the testing.

It is stated that insufficient data are available on the performance of small orifices. When applied to carburetors it is found that the same-bore orifice would change the metering characteristics, depending on temperature of the fuel, design of the orifice with respect to its length of bore, area of approach, discharge, and various other factors.

The article is devoted mainly to a description of the experimental equipment and states that at the present time data are available on 56.5 Be. gasoline, 41.2 Be. kerosene, and distilled water. It also gives a chart showing actual discharge and effect of temperature on orifice discharge with gasoline, the orifice diameter being 0.030 in. and the thickness of orifice plate 0.5 in., with the discharge plotted against fuel temperature and head on orifice. (*The Automotive Manufacturer*, vol. 64, no. 8, Nov., 1922, pp. 7–8, 3 figs., *e*)

FARMAN AVIATION MOTOR, 600-HP. 18-CYLINDER W-TYPE. In this motor the cylinders are 130 mm. (6.1 in.) bore by 180 mm. (7.1 in.) stroke, located in three rows 40 deg. apart. This angle of 40 deg. was determined with a view to having a steady torque, the motor giving 18 impulses in each two revolutions or 720 deg., which gives one power stroke for each 40 deg.

The crankshaft has six throws. The use of a W-type motor was considered to be of advantage as it reduces the size of the crankcase. The increase in the number of cylinders leading to a reduction of the unit cylinder volume permits of a higher compression without fear of self-ignition owing to heating of the center of the piston. The weight of the motor with its full equipment of ignition carburation pumps, etc., is only about 780 kg. (1716 lb.) for an effective horsepower of about 800 at 2200 r.p.m. (The nominal rating of the motor is 600 hp. at 715 r.p.m.)

The motor drives the propeller through a reducing gear weighing 45 kg. (99 lb.). In addition to this there is a 1200-watt generator with a wireless-telegraph alternator weighing 34 kg. (86 lb.), and the total weight of the motor with all accessories including propeller hub is 920 kg. (2025 lb.). The cylinders are grouped in pairs and the compression ratio is one to six.

The cylinders are made from forgings and welded together in pairs. Each pair has a common water jacket made of sheet steel and welded over the group. There are four valves per cylinder, these valves being made of high-nickel steel. Each valve has two concentric springs wound in opposite directions and each cylinder carries three spark-plug bosses, two side by side and the third 180 deg. therefrom; this third boss is intended for the insertion of starting devices, be they explosive cartridges or compressed-air plugs. The article describes in some detail the design of such parts as crankshaft, pistons, valve gear, ignition and lubrication. In connection with this latter it is of interest to note that the oil filters are so arranged that they can be inspected while the engine is running. A safety valve is provided on the oil circulation pump to open when the oil pressure becomes excessive. The purpose of this is to prevent the breaking of the oil radiator at starting in cold weather when the oil is very viscous. Four Zenith carburetors are provided, two with single outlets and two double ones, each outlet feeding three cylinders. The carburetors are fed by two A M fuel pumps connected in parallel to each pump, being, however, of sufficient capacity to supply fuel to all the four carburetors. There is a device for setting these pumps in operation automatically as soon as the electrical starter begins running, in addition to which the pumps may be also started by hand. As it would be impracticable to start a motor of this size by hand, an electric starter has been provided. (*L'Aéronautique*, vol. 4, no. 42, Nov., 1922, pp. 343–347, 8 figs., *dA*)

Small British Two-Stroke Cycle Marine Engine

BRUNTONS TWO-STROKE MARINE ENGINE. Description of an engine built by Bruntons, of Sudbury, England, in which some novel features are embodied. The cylinder diameters are $2^5/_8$ in. bore and the stroke is 3 in. On brake test 6 hp. was obtained when running at 1000 r.p.m. which was raised to 8 hp. at 1400 r.p.m. The engine is illustrated in Fig. 1. The pistons are of aluminum alloy and are provided with two cast-iron piston rings at the top and a third one at the bottom. The connecting rods are made of nickel-chrome steel and are fitted with phosphor-bronze bushes to receive the gudgeon pins, and with roller bearings at the big ends.

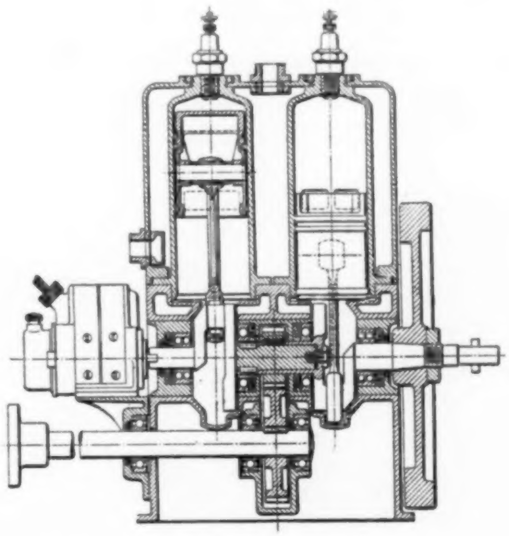

FIG. 1 BRUNTONS TWO-STROKE-CYCLE SMALL MARINE ENGINE

A double-helical toothed wheel on the crankshaft, at about the center of its length, engages with another wheel on the lower shaft, for which a speed one-half of that of the crankshaft is obtained. The crankshaft runs in ball bearings. To prevent the air leakage that would otherwise take place an effective oil packing is arranged with the use of a phosphor-bronze casting with concentric grooves cut in it. The speed reduction gearing is kept completely isolated from the two separate crank chambers. A positive gear pump which is driven by a chain from the propeller shaft, is used to circulate the water required for cylinder cooling. Magneto ignition complete the equipment of what constitutes a complete power unit. The endeavor to produce a small two-stroke unit under conditions suitable for obtaining high efficiency and at the same time with a speed for the driving shaft which is suitable for the use of a reversible propeller, has engaged the attention of engineers for a considerable time, and this installation is not lacking in interest to all who have been faced by this problem. Among other engines described, particular attention is called to the Kelvin engine with single-sleeve valves built under the Burt-McCullum patents. The single sleeve has a motion which is a combination of a reciprocating vertical movement and a turning action upon the sleeve. Double inlet and exhaust ports register with ports leading through the cylinder walls, permitting entry and exhaust of the gases. (The Marine and Small Craft Exhibition. *Engineering*, vol. 114, no. 2969, Nov. 24, 1922, p. 645, 3 figs. Complete article p. 644–647, 10 figs. *d*)

MACHINE PARTS

Reversing Gear for Marine Turbine

Epicyclic Reversing Gear for Ljungström Marine Turbines. The interest in this transmission—which is not new in itself—lies in the fact that it is applied on a turbine-driven steamer of 2150 aggregate effective horsepower. The boat in question, the *Pacific*, which is owned in Denmark, is of 7400 tons dead weight, and is driven by a Ljungström steam turbine running at 3000 r.p.m. and designed to operate with steam supplied at a stop-valve pressure of 185 lb. per sq. in. and a temperature of 662 deg. fahr.

The Ljungström turbine has two intermeshed rotors turning in

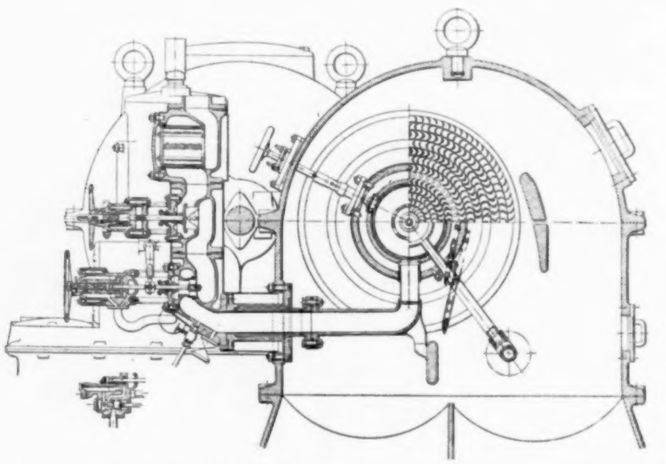

Fig. 2 General Arrangement of the Ljungström Epicyclic Reversing Gear for Marine Turbines

opposite directions and it is always mounted on top of its condenser.

The general arrangement of the gearing is shown in Fig. 2, which represents diagrammatically a plan (partly in section) of the turbine first transmission shaft, reverse gear, the second reduction gear, and the thrust bearing. Each rotor shaft carries a pinion which

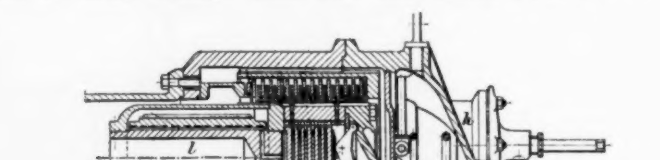

Fig. 3 Detailed View of Thrust Bearings and Lubrication in Ljungström Epicyclic Reversing Gear for Marine Turbines

drives a double-helical gear wheel on the first transmission shaft. As the two rotors turn in opposite direction an idler (not shown) is interposed between the rotor pinion and the gear wheel. In this first stage the speed is reduced from 3000 r.p.m. to 540 r.p.m., which is further reduced to 70 r.p.m. in the second reduction stage shown to the right of the reverse gear.

The details of this reverse gear are illustrated in the original article, which also contains several diagrams together with a detailed description of the operation of the gear.

The oil for lubrication purposes enters at the port *k*, shown on the left, and as it escapes is collected in the fixed housing and conveyed away for filtration, after which it is again circulated through the system. The wheels *a* and *d* are clearly shown to the left of Fig. 3, and as will be seen, a ball thrust bearing is interposed between the opposing ends of the two shafts. This bearing has, it will be seen, an elastic seating on the left-hand side, which consists of a slightly dished steel ring similar to a component of a Belleville spring. Similar elastic cushions, it will be noted, are provided at each end of the outer multiple plate clutch. The planet pinions are mounted on hollow steel studs as shown at *l*, Fig. 3. They are themselves lined with white metal, and the bearing is supplied with oil under pressure through the ports shown to the right. Of each pair of pinions, one has teeth covering half its length only, as is best seen near the bottom of Fig. 3. This toothed portion gears with the wheel *a* and with its fellow-pinion which, as shown at the top of Fig. 3, has teeth extending over its full length and gears with the wheel *d*.

It will be seen that with this arrangement for reversing there is no necessity for stopping the turbine when maneuvring, and the gears are so well lubricated that even continuous slipping of the clutches causes little rise of temperature. Nevertheless provision is made by which in maneuvering the turbine speed is automatically reduced to 50 per cent of its normal value; the maneuvering wheel is fitted near the stop valve and is coupled up to the steam regulating valves so that during a change over the steam is partially shut off. A diagram showing the alterations in the turbine and propeller speeds during a change over shows that the maximum temperature rise did not exceed 20 deg. cent.

The T. S. *Pacific* has been twice to Australia, and has since returned to Denmark from a voyage to Honolulu. The reports from the owners are highly satisfactory. The total consumption of coal per day has been on an average 23 tons for all purposes with the turbines developing about 2300 i.hp., or 0.92 lb. of coal per i.hp. per hour. Average bunker coals were used, and the percentage of ash was on an average 16½ per cent. When burning oil the consumption has been 16 tons per day.

A second vessel, the *T. S. Stal*, has also been fitted with a turbine of the same type developing 1350 i.hp. In this case the reverse gear is placed abaft the main-gear casing in order to increase its accessibility. (*Engineering*, vol. 114, no. 2971, Dec. 8, 1922, pp. 699–703, 24 figs., *dA*)

MACHINE SHOP

Hob Making and Special Cutter-Grinding Machinery

Simmons Method of Hob Making, Charles O. Herb. In the Simmons method a standard worm thread is first milled on the

Fig. 4 Generating and Relieving Hob Teeth by Employing a Revolving Cutter and a Standard Relieving Attachment

hob in a thread-milling machine by means of a straight-sided formed milling cutter. After heat treating this preliminary thread is "generated" to the true tooth form in the relieving lathe by the action of a rotating cutter somewhat similar to that employed on the Fellows gear shaper. Because of the action of the cutter the teeth are of the true involute form and mesh with standard gears of the same diametral pitch. The front face of the cutter teeth has a rake of about five degrees.

The two operations of particular interest in this process are the relieving operation which is performed in a lathe which, in addition to the standard relieving attachment, is equipped with a special attachment mounted on the carriage, and the grinding of the cutter teeth done in a machine designed for this purpose. The special attachment used in performing the relieving operation (Fig. 4) consists essentially of a cutter head *A* mounted on a special cross-slide of the lathe and a unit attachment to the front of the carriage for transmitting the rotary motion to the cutter. Power is delivered to this unit by means of a gear on the headstock spindle directly in back of the faceplate which meshes with idler gear *H* on bracket *B*. This idler gear, in turn, drives a gear *K* on the forward end of the large screw *C*, which is also designated by the letter *C* in Fig. 5.

From this screw power is transmitted through worm wheel *D* and sliding shaft *E*. At the cutter-head end of the sliding shaft is an integral bevel gear which meshes with a second bevel gear on the lower end of cutter shaft *F*. By designing shaft *E* to slide through worm wheel *D* the cutter may be moved toward the work at the same time that the carriage is operated along the ways of the

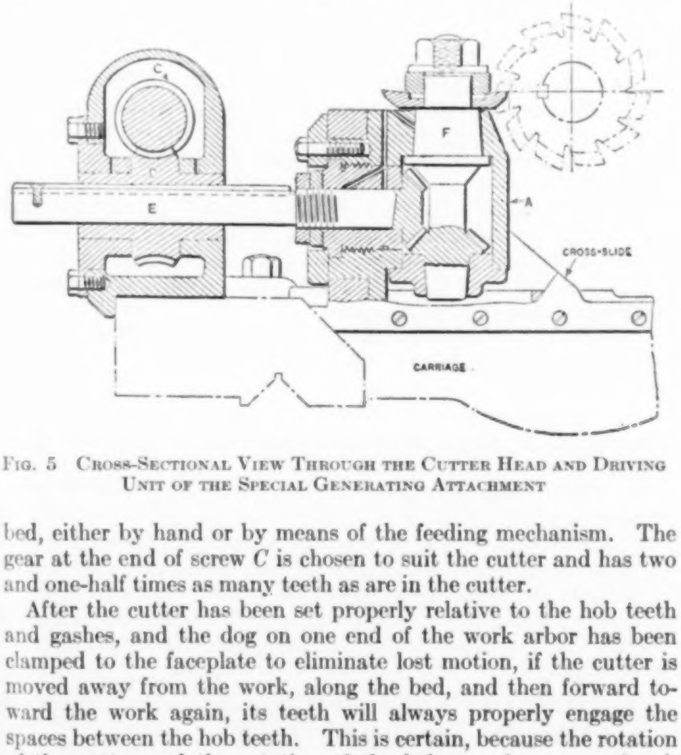

Fig. 5 Cross-Sectional View Through the Cutter Head and Driving Unit of the Special Generating Attachment

bed, either by hand or by means of the feeding mechanism. The gear at the end of screw *C* is chosen to suit the cutter and has two and one-half times as many teeth as are in the cutter.

After the cutter has been set properly relative to the hob teeth and gashes, and the dog on one end of the work arbor has been clamped to the faceplate to eliminate lost motion, if the cutter is moved away from the work, along the bed, and then forward toward the work again, its teeth will always properly engage the spaces between the hob teeth. This is certain, because the rotation of the cutter and the rotation of the hob are always correlated. When the carriage is moved along the bed, the cutter rotates faster than when the carriage is stationary, due to the rolling of worm wheel *D* on screw *C*.

The cutter for producing gear hobs is inclined so that as each tooth engages the work, the face is in a plane at right angles to the helix of the hob thread. The hob teeth are therefore "generated" rather than "formed." The cutters are made from forged disks. These disks are rough-drilled, rough-faced on both sides and rough-turned in a lathe, given an annealing and then permitted to season. The next step consists of finish-boring, finish-facing both sides, and finish-turning the peripheral surfaces to the required angles.

The cutter disks then undergo several machining and grinding operations, of which the most interesting is that of grinding the tooth profiles. For this purpose a special machine described in the original article is used. (*Machinery*, vol. 29, no. 4, Dec., 1922, pp. 255–260, 11 figs. *d*)

MARINE ENGINEERING (See also Machine Parts)

Discarding Geared Turbines. It is stated that several ship owners are removing geared turbines and replacing them by reciprocating engines. Two American-built steamers, the *Wildomina* and *Sylvia Victoria*, which were fitted with geared-turbine machinery are in the hands of a Clyde firm which is substituting for it ordinary triple-expansion reciprocating engines. Other ship owners are said to be showing a disposition to revert to reciprocating engines for their new vessels. The 11,300-ton steamer *Port Hardy*, for the Commonwealth and Dominion Line, recently launched, is to be fitted with twin-screw triple-expansion machinery developing 4500 i.hp. It is said that the *Port Hardy* had really been designed for geared turbines, but there had been a great deal of trouble with the double-reduction-geared turbines and as the line already had seven vessels with turbines they preferred not to continue with that type of engine until they had gained a little more experience. Quite possibly they would return to that type at a later date. The last ship, the *Port Hunter*, had run trials on turbine engines which had proved to be among the smoothest they had giving great hopes for the future. (*Practical Engineer*, vol. 66, no. 1866, Nov. 30, 1922, p. 344, *g*)

METALLURGY

New Type of Open-Hearth Furnace. Description of the Loftus furnace, a producer-gas furnace wherein the producer gas is discharged from a restricted nozzle into a mixing chamber serving to induce the necessary air for combustion. A booster supply of regenerated air is withdrawn from the regenerative chamber, passed through a special chamber driven by a variable-speed motor and introduced into the center of the stream of gas in the direction of its flow through the furnace. The pressure and volume of this booster supply of regenerated air may be governed by the control of the variable-speed motor, in this way giving the operator control over the flame. With such a furnace construction more efficient combustion is obtained, as shown by the actual operation through the mixing of the air with the gas in the hot mixing chamber. The air and gas enter an extremely hot mixing chamber in the form of an outer envelope of air completely surrounding an annular ring of gas containing an inner core of air which is calculated to produce an extremely intimate mixture. This mixture is discharged from the mixing chamber in the form of a flame having practically no stratification and with the utmost compactness and continuity of construction. The operator can control the flame by regulating the pressure and flame of the booster supply of regenerated air. If he wishes he may melt down a charge with a short hot flame and refine with a long mellow flame which completely covers the bath without extending through the outgoing channels.

The original article describes and illustrates the furnace in some detail. Such a furnace has been operating on producer gas at the Ohio Works of the Sharon Steel Hook Co., Lowellville, Ohio, since June 8, 1922. While the charging conditions were not ideal the furnace is said to have produced an average of approximately 30 per cent more steel than the other furnaces in the plant on approximately 30 per cent less coal per ton of ingots. The furnace, up to December, 1922, has given more than 300 heats and from all indications it is believed it will run about 400 heats or more, making an increase of about 50 per cent in its usual life. The repairs have been almost negligible as it has been possible to get almost 100 heats from back and front walls and approximately 80 heats off the gooseneck. (*The Iron Age*, vol. 110, no. 26, Dec. 28, 1922, pp. 1677–1679, 5 figs., *d*)

Ingot Corner Segregation in a Nickel-Chrome Steel, T. Henry Turner. In investigating by means of sulphur prints an octagonal chrome-nickel steel forging the author discovered the presence of planes of segregation. Such planes were found at the angular portions of the ingot and represent the meeting of crystals which have grown from the cooling faces into the still liquid metal. The author believes that such forgings should be rejected even though the mass of the material may possess good chemical and mechanical properties, as the defects, consisting in planes of segregation, entirely negative these good properties and make the forging as a whole treacherous and unreliable in the extreme.

The author makes the following recommendations:

1 That the bell-shaped cylinder design should never be used unless the supplier can guarantee—

a Absence of planes of segregation in the material

b Adequate working during the forging operations.

2 That in cases where the supplier cannot guarantee adequate working of the material during forging (owing to the nature of the design and the limitations imposed upon his work by the nature of his available tools), a composite ring made up of two or more rings, which can be manufactured by a satisfactory forging or rolling, would probably prove more reliable.

3 That the final rolling of such rings is to be preferred to forging them throughout, in that it would tend to produce greater uniformity of dimensions in the rough, and therefore economy in machining. Also it should produce a better and more uniform internal structure and greater uniformity of mechanical properties.

4 That in the case of all large forgings which will be subjected to appreciable circumferential loading, the supplier should be asked to face up both ends of the block of steel before forging is commenced. Sulphur prints should then be taken and forging only proceeded with after these have been passed as satisfactory. Such an operation would, of course, be costly, but it would only be advisable when appreciable hoop stresses are expected.

5 That in cases where the steel maker considers recommendation No. 4 impracticable, the elimination of ingot corner segregation may be achieved by forging down octagonally and machining off the corners which would still contain the segregation. This suggestion is perhaps the most useful and that most likely of successful adoption.

6 That the centrifugal casting of steel for such large cylinders or rings, and even for large turbo-generators rotors, is feasible and worthy of serious consideration. (*Engineering*, vol. 114, no. 2969, Nov. 24, 1922, pp. 662–664, 9 figs. Compare also abstract in *Iron Trade Review*, vol. 71, no. 25, Dec. 21, 1922, pp. 1695–1700, 5 figs. *dA*)

MOTOR-CAR ENGINEERING

Chevrolet Copper-Cooled Car, J. Edward Schipper. Description of an air-cooled car the particular feature of which is that it employs copper fins electrically welded to cast-iron cylinders.

To manufacture the copper-cooled engine it has been necessary to develop machinery to form the copper ribs, and apparatus to handle commercially the electric welding of the formed ribs to the cylinders. The copper fins are made in a special machine which takes the sheet copper and forms it into the proper shape, reducing the depth of the fins or loops at points where clearance for the valve push rods and adjacent cylinders is required. The copper is so formed that the loops are closed at points adjacent to the cylinder barrel, and in such a way that a continuous band of copper is in contact with the outer wall of the cylinder, which latter is machined both inside and out.

The cylinders are manufactured singly but are assembled in pairs. Between the cylinders in each pair, which are closely adjacent to one another, the fins are graduated in size so that there is no interference.

The cooling is by air forced through and around the fins and over the cylinder heads by a centrifugal blower made largely from sheet aluminum alloy. This blower is in front of the engine and is driven by a special form of V-belt from a pulley on the crankshaft. Air enters at the bottom of the fins around the cylinders. A light steel stamping forms a tubular draft tube which fits closely around the cylinders so that air can enter the fan only by passing upward through and around the copper fins and through the space above the cylinders. The V-belt employed has a woven-cord construction on its neutral axis and a durable elastic material on either side of the neutral axis permitting of the flexing necessary for the drive. The drive for the fan is triangular, the belt passing around the crankshaft fan and generator pulleys. Adjustment for the tension of the belt is secured by swinging the generator, which is mounted on the left side of the engine. The engine, it may be noted, is the first "square" one to be placed on the market for some time, both bore and stroke being $3^1/_2$ in. With a compression ratio of four to one it develops a maximum brake horsepower of 22 at 750 r.p.m. (*Automotive Industries*, vol. 47, no. 26, Dec. 28, 1922, pp. 1259–1265, illustrated, *d*)

PHYSICS

Temperatures at Which Physical Changes Occur, Henry D. Hibbard, Mem. Am.Soc.M.E. A list of some of the temperatures relating in one way or another to the production and heat treatment of steel and to the fusibility of steel and of its components. Of these the following, all belonging to the higher ranges, are of interest:

Degrees Temperature Cent.	Fahr.		Degrees Temperature Cent.	Fahr.	
1750	3182	Pure silica melts	2220	4028	Carborundum decomposes without melting
1755	3191	Platinum melts	2450	4442	Iron boils
1795	3263	Titanium melts	2500	4532	Zirconium oxide melts
1820	3308	Bauxite melts	2550	4622	Molybdenum melts
1900	3452	Manganese boils	2572	4662	Lime (CaO) melts
2050	3722	Chromite bricks melt	2750	4982	Vanadium carbide melts
		Pure alumina melts	2800	5072	Magnesia (MgO) melts
2059	3738	Chromic oxide (Cr_2O_3) melts	3176	5750	Tungsten melts
2165	3929	Magnesia brick belts	3600	6512	Molybdenum boils
2180	3956	Chromite melts	3700	6692	Carbon is vaporized
2200	3992	Chromium boils	5200	9392	Tungsten is vaporized
		Silicon boils			
		Aluminum boils			
		Graphite made from amorphous carbon (2200 to 2400 deg.)			

An interesting discussion is also submitted as to the proper definition of the melting point of a substance and the temperatures near the melting point which bear some relation to it. Some of these are practically useful and some have only a scientific value. These are:

1 The temperature at which the substance sinters, as a clay. This is useful. It is below the fusion point, No. 4.

2 The highest temperature at which a vessel made of the substance can be used. Instance a porcelain crucible. This is useful. It is lower than No. 1.

3 The temperature at which it softens in any degree which, as in a firebrick, limits its usefulness. This is useful. It is lower than the fusion point, No. 4.

4 The temperature at which its edges are rounded. This is often called the fusion point and is generally the one given in the table. It has only a scientific value.

5 The temperature at which it becomes fluid. Instance a slag. This is interesting, but taken alone is not useful. It is above No. 4.

6 The lowest temperature at which it may be handled molten in practice, as a slag or metal in a metallurgical operation. This is useful. The temperature may be 50 or 100 deg. cent. (90 to 180 deg. fahr.) above No. 5 as the substance leaves the furnace (*The Iron Age*, vol. 110, no. 23, Dec. 7, 1922, pp. 1492–1493, *g*)

POWER-PLANT ENGINEERING (See also Engineering Materials; Fuels and Firing)

Boiler-Tube Failures

Boiler-Tube Failures, Haylett O'Neil. The author claims that in a modern boiler plant, well designed and well operated, that is, where the boiler is clean and proper internal circulation is maintained, boiler-tube failures are infrequent. There may occur, however, conditions under which boiler tubes fail, and he describes one such case which gave considerable trouble.

In general, boiler tubes may fail where there is a condition producing sufficient unevenness of temperature to cause rates of expansion in adjacent parts of a tube varied enough to strain the metal. In a case described by the author it was found that little trouble was experienced from boiler-tube failures in the spring and summer, but the tubes would fail rapidly in the late fall and winter. An investigation showed that excessive scale was deposited from water during these two latter periods, no water softening being used. Furthermore, during the dry season feedwater was drawn

from a spring in which the water was charged heavily with bicarbonate of lime. Careful records presented in the form of curves in the original article show a clear relation between the use of spring water and tube failure. Scale deposited on the tubes would give a uniform heat insulation, therefore, blisters would not occur, but the tubes cracked square off at the front end in nearly all cases.

In the ordinary vertically baffled boiler the author has noticed that under similar scale conditions the tube generally bows down. This indicates that the under side of the tube has been heated hotter than the upper, which is reasonable with this arrangement of baffling, and hence has bulged toward the fire.

In a clean boiler the temperature of the metal is practically equal to that of the adjacent water, but scale insulates the metal of the tube from the water. In that case the hotter the gas is the hotter will be the metal, irrespective of the water temperature. How, then, does the top of the tube get hotter than the bottom? The answer in one case proved to be that the T-tile kept the top of the tube hot during both normal operation and part of the cleaning periods. The temperature of this tile according to the pyrometer was close to 2500 deg. fahr. The lower part of the tube was surrounded by furnace gas which fluctuated between 2500 deg. during firing conditions and about 1200 deg. during cleaning periods. Inasmuch as during a cleaning period the front doors had to be open, temperature variations could not be eliminated.

On account of the heat conductivity of the iron it would naturally be impossible to maintain for any considerable time a temperature difference between the top and bottom of the tube, but calculations have been made by the author indicating that only a few hundred degrees difference is required to curve an 18-ft. tube as much as 3 in. out of line. Such a temperature difference seems to be entirely feasible.

What actually happened is this: At first a tube would be humped upward slightly during a cleaning period and restraightened afterward as soon as the bottom could be expanded under the influence of the furnace temperature. Every time cleaning or banking took place the expansion and contraction cycle was repeated. Therefore the tube was subjected to repeated bending upward and downward several times a day. The strains were concentrated at the tube sheet, where finally the tube became brittle and lost its tensile strength. Finally, the tube would be strained beyond its elastic limit and take on a permanent set. When the boiler was cooled for cleaning the temperature was reduced nearly to room temperature, causing extreme tension in the tube due to contraction, and the tube would break.

The final solution of the trouble came in the regular adding to the boiler water of an internal treatment, the base of which was caustic soda. A given quantity of this was pumped into each boiler at regular intervals, and twice each week boiler waters were titrated for soda concentration. By keeping the soda concentration under 40 grains per gallon, excellent results were obtained. After a run of from 60 to 90 days on spring water only a thin film of scale was formed, and there was no foaming. The T-tile were finally replaced entirely.

Actual records of tube replacements have shown that this manner of handling the trouble was correct, and certainly no steam-power plant is even approximately safe unless the boilers are kept clean religiously. (*Power*, vol. 56, no. 23, Dec. 5, 1922, pp. 876–878, 7 figs. *pe*)

Tests of a Magazine-Feed Boiler with Special Method of Supplying Secondary Air, John Blizard, J. Neil, and A. Pincus. Data of steaming tests carried out by the Fuel Section of the U. S. Bureau of Mines at the request of the American Society of Heating and Ventilating Engineers on a magazine-feed boiler provided with special method of supplying secondary air to determine its thermal efficiency and other important factors when fired with various fuels.

It was also found that coke, anthracite, lignite char, and various bituminous coals could be burned with little poking and other attention to the fuel on the grate or in the magazine except when cleaning the fire or charging fuel, but that caking Pittsburgh coal was unsuited to this boiler since it had to be poked frequently to break up an arch which formed near the top of the grate.

The outstanding results of the tests were: That the method of admitting secondary air supply over the fuel bed permitted the combustible gases rising from the coke fuel bed to be burned so that only 29 to 58 per cent excess air remained after combustion, those from anthracite with only 35 to 62 per cent excess air, those from Illinois coal with 61 per cent excess air, and those from Colorado coal with only 23 per cent excess air. With all of these fuels the carbon monoxide left in the gas represented only about 1 per cent or less of the heat of the fuel. Therefore, for these fuels the methods of admitting secondary air was good. On the other hand, the carbon monoxide remaining in the gas when burning Pittsburgh coal with 35 per cent excess air represented 7 per cent of the heat energy of the coal, and the unscreened lignite char was burned with 221 per cent excess air, which was reduced in the trial with the screened char to 94 per cent.

The thermal efficiency with coke was highest when operating at about 90 per cent of the maker's rating; at this rating the efficiency was 76 per cent, and dropped to 67 per cent and 70 per cent when operating at 123 and 35 per cent rating, respectively.

Data have also been obtained for other kinds of coal, such as stove and chestnut-size anthracite, briquets, and Pittsburgh coal.

An interesting series of supplementary investigations were also carried on and temperatures above the fuel bed and in one of the central flues were measured; likewise temperatures in the flues of the first and second pass.

Among other things, it would appear that at the highest rate of steaming and the lowest rate but one, the furnace temperatures with coke and anthracite are approximately the same, while at the highest rate but one the temperature was higher with anthracite than with coke by 150 deg. fahr. and at the lowest rating the anthracite furnace temperature was 350 deg. fahr. lower.

Data for other temperatures are given in the original article. (*Journal of American Society of Heating and Ventilating Engineers*, vol. 28, no. 9, Dec., 1922, pp. 833–842, 5 figs., *e*)

Electric Drive in a Sugar-Manufacturing Plant, Roger B. Stevens. Description of the layout of electrical equipment of the recently completed Baltimore plant of the American Sugar Refining Co., the particular feature of which is that practically the entire drive is effected by means of direct-current motors.

There were several conditions which decided in favor of direct current. In the first place the sugar refinery consists of a number of closely grouped buildings with the transmitting distances short and the loads closely concentrated, which make transmission losses small. Furthermore, the conditions of sugar-refinery work are somewhat peculiar. The process from start to finish is continuous. Moreover, the materials are carried at a fairly high temperature and even a small delay leading to loss of heat produces stiffening of the material. Because of this the power required to start a given machine may be from five to ten times that required for operation after the running condition has been established. Nearly all of the refining processes are interdependent and continuous, so that unless large and expensive storage facilities between the various stages are available the several elements of machinery must be capable of close adjustment in speed. Furthermore, in the case of more than 80 per cent of the total horsepower required, either a speed adjustment was absolutely necessary or a very distinct advantage was to be gained by providing such adjustment.

In alternating-current machinery only the slip-ring motor with resistance is capable of an extensive range of speed variation, but it is then only the equivalent of a series-wound direct-current motor and has the disadvantages of moving contacts and vulnerable brush rings. The direct-current motor, on the contrary, is capable of good starting torque and adjustable speed, though it is slightly higher in the first cost. Furthermore, with the close grouping of the loads the comparatively low (230) voltage of the direct-current made the total cost of the distributing system lower than would have been the case with alternating current. This is principally because a large number of motors could be grouped on single feeders to take advantage of the low diversity factor thus obtainable.

The article describes in detail the installations used and the electrical distribution system. (Factors in Industrial Plant Distribution. *Electrical World*, vol. 80, no. 24, Dec. 9, 1922, pp. 1259–1262, 5 figs., *d*. Compare also editorial entitled, An Unusual Case in Industrial Power Design, same issue, p. 1254)

Exploding an Old-Time Boiler Fallacy, W. B. Roberts. Data of tests to show the fallacy of the belief that cold water sprayed upon a red-hot boiler plate instantly generates an uncontrollable volume of steam.

The article is based on tests carried out under the auspices of the Manchester Steam Users' Association on an ordinary Lancashire boiler 7 ft. 0 in. diameter and 27 ft. 9 in. long, shell plates $^7/_{16}$ in. thick and head plates $^9/_{16}$ in. thick. A number of precautions were taken to protect observers from the consequences of the expected explosion and, at the same time, permit them to take the necessary observations. The tests were carried out in such a manner that at 6.30 p.m. there was 25 lb. of steam pressure showing and the safety valves began to blow. At the same time the blow-off top was opened full bore, there being $9^1/_2$ in. of water covering the crown of the furnaces. Within about an hour the level of water was brought to $3\,^5/_8$ in. below the crown of the furnace, this latter laying bare and exposing a strip of plating 21 in. wide at a temperature of about 700 deg. fahr. with a pressure of $28^1/_2$ lb. on boiler, and the safety valve blowing freely. The feedwater having a temperature of 60 deg. fahr. was then turned on, sprayed directly on the red-hot plates with a rate of injection a little more than $4^1/_2$ cu. ft. per min. Showering the feedwater at a temperature of 60 deg. fahr. on a red-hot furnace crown was not attended by an increase of pressure. On the contrary, the pressure at once began to fall and in $2^1/_2$ min. had fallen from $28^1/_2$ lb. to 26 lb.

Upon looking into the furnaces the ring seams of the furnace tubes were found to be leaking freely. The overlaps of the first seven ring seams in each furnace were sprung at the crown, but the shape of the furnace tubes was practically unaltered except that the right-hand tube was slightly bulged in three places.

Other slight consequences of overheating were observable. The main thing, however, which the tests have shown is that spraying fairly cold water on red-hot plates and furnaces did not lead to the ripping of the plates circumferentially or longitudinally, and did not cause any increase of pressure due to a violent and sudden generation of steam. (*Power House*, vol. 15, no. 24, Dec. 20, 1922, pp. 19–21, 5 figs., *eA*)

Discussion of Vapor Refrigerating Condensers for Steam Turbines, F. C. Evans. One of the means of bettering the economy of the steam turbine is to reduce the back pressure. At the present time the back pressure employed is limited to around 1 in. of mercury absolute, corresponding to a temperature of the exhaust steam of about 79 deg. fahr. This limit has been reached with water as the condensing medium, and the author questions whether better results could not be obtained by substituting the cooling coils of a refrigerating system for the condensing-water tubes of the ordinary surface condenser, so as to reduce the temperature of the exhaust steam to about 35 deg. fahr.

For purposes of discussion the author considers a plant where steam is available from the boilers at 200 lb. per sq. in. absolute pressure and 200 deg. superheat, and ample condensing water at a temperature of 60 deg. fahr. but which must be raised 20 ft. He shows that under these conditions the available energy in the steam is 413 B.t.u. per lb. of steam less 1.5 B.t.u. required to handle the water, which gives a net energy for sale of 411.5 B.t.u. per lb. of steam.

When a vapor refrigerating condenser has been substituted for a water condenser, the available energy in the steam operating with the back pressure of 0.205 in. is 486 B.t.u. per lb. But the author shows that the refrigerating system consumes 90 B.t.u., thus giving as a final result 396 B.t.u. of energy available for sale per pound of steam, which is less than when a water condenser is used.

As a result of this analysis it appears that at the present time steps o increase the economy of the steam turbine must be taken in another direction. What this direction may be one cannot say, but it seems impossible to reduce the back pressure of the turbine economically by the use of a vapor refrigerating condenser. (*The Sibley Journal of Engineering*, vol. 36, no. 9, Dec., 1922, pp. 175–177 and 187, 3 figs., *tg*)

Steam and Steam Boilers

Steam and Steam Boilers. An editorial based on Mr. C. E. Stromeyer's report to the Manchester Steam Users' Association.

Among other things the report includes a partial description of some interesting experiments on the strength of very old boilers. Of these one was obtained from Messrs. Pilkington and had been constructed some sixty years ago. The other, supplied by the Metropolitan Water Board, was of somewhat similar date. The experiments are still incomplete, but they indicate pretty conclusively that the material of the shell plates has not deteriorated with time, and that injuries and cracks caused by the punching of the plates had not apparently extended during the very long life of the boilers. Specimens of the material tested showed somewhat erratic results, but it is probable that this was characteristic of the iron made at the period in question. The range of tensile strength was from 17 tons to 25 tons per sq. in. In many of the seams, cracks due to punching extended over several rivet holes, but these do not appear to have had any worse effect than that of reducing the net section of the metal. These observations seem to indicate that factors of safety might reasonably be reduced now in view of the uniformity of the material supplied, the great care exercised in its manufacture and the prevalent practice of drilling all rivet holes.

During a hydraulic test one of the boiler shells "barreled" as first described by Mr. J. C. Spence in a paper read before the North-East Coast Institution of Engineers and Shipbuilders in 1891. On making measurements on a model boiler tested to destruction Mr. Spence found that the shell stretched less at mid-length than it did at a certain short distance from the end plates. The observation was novel at the time, and the curious bulges in question received the name of "Spence humps."

Mr. Stromeyer states that it has generally been thought that the effect of the end plates was to relieve the circumferential stress. This may well be true, but there is no excuse for such misunderstanding for, in the discussion of Mr. Spence's experiments, which appeared in *Engineering*, Apr. 27, 1891, p. 468, it was shown that save in the case of extremely short boilers the effect of the ends was to cause a local increase of the circumferential stress. This was very clearly brought out in Mr. Spence's experiments, and was fully confirmed by the mathematical theory developed in a paper by Mr. (afterwards Professor) J. T. Nicolson, read at the same time as that of Mr. Spence. The subject was very fully discussed in *Engineering*, both editorially and in its correspondence columns, and that the other view should, as Mr. Stromeyer states, still be prevalent is not creditable to boiler designers and constructors.

In another section of his report, Mr. Stromeyer suggests that very slow alternations of stress are more serious than rapid ones. It appears quite possible that in the case of exceedingly rapid repetitions of stress the metal may fail to take the full deformation theoretically due to the load, since, with all materials, when a load is first applied the full strain is not simultaneously achieved. There is always some "after creep" and it is thus quite possible that the metal is less severely tried by very rapidly alternating stresses than by slowly applied ones. The general view has, however, been that at any ordinary rates of alteration the fatigue limit of our structural materials is independent of the rate of repetition. Mr. Stromeyer's view is, however, that the very slow rate of changes of stress to which a boiler is subjected, makes these much more injurious than if the variations were more rapid. This is contrary to the common belief and does not seem altogether consistent with the fact that slightly overstrained metal in time recovers its elastic properties. In support of his theory, Mr. Stromeyer instances the grooving found at the roots of the flanges of dished ends, but grooving is only indirectly connected with the stress on the metal as it is primarily an oxidation process, facilitated by the differences of E.M.F. which arise between strained and unstrained metal. It seems to us questionable, therefore, whether fatigue in the usual understanding of the term is involved in the matter any more than it was in the extraordinary failures of the high-tension bronzes adopted for the Catskill aqueduct. (*Engineering*, vol. 114, no. 2973, Dec. 22, 1922, pp. 775–776, *g*)

RAILWAY ENGINEERING (See also Engineering Materials)

Recent Tendencies in British Locomotive Practice. E. C. Poultney. Beginning of a series in which it is proposed to describe a number of typical recent locomotives built for service

on British railways which indicate the tendencies of modern practice.

Perhaps one of the most striking features is the use made of either three or four cylinders, with the three-cylinder arrangement rapidly gaining favor. (The ordinary British type of locomotive has two inside cylinders.)

Steam pressures have undergone some changes during recent years. Previous to the general introduction of flue-tube superheaters, boiler pressures ranged from about 170 to 200 lb. With the advent of superheating there was a general tendency to reduce pressures to about 160 lb. and increase the cylinder volume. The practice is now toward the higher pressures, and those of 180 to 200 lb. are considered necessary, while cylinder volumes are continually increased to furnish the necessary power; in fact, it is largely the greater cylinder capacity now required that has prompted the introduction of three- and four-cylinder locomotives.

Locomotives now built for main-line working are practically all fitted with flue-tube superheaters, and older engines when requiring extensive repairs and renewals, or to be rebuilt, are as a rule superheated; and when this course is adopted it is usual to fit new cylinders with piston valves. While the type of superheater used is in all instances the same, there are in some cases considerable differences in details. This applies more especially to the design of the headers.

Damper gear has been discarded now by all railways. Automatic air valves are quite often fitted to the steam chests and sometimes on the header (the wet-steam side). Cylinder by-pass valves are not generally used, neither are pyrometers.

From this the author proceeds to describe some of the new locomotives, and in connection with the Great Central 4–6–0 type describes and illustrates the Robinson piston and pressure-relief valve.

An interesting fitting used on the Great Central is what is called an Intensifore lubricator which works in much the same manner as a hydraulic intensifier. Briefly, the arrangement consists of a container filled with oil upon which pressure is applied by means of steam acting as a plunger. Oil is led from the lubricator by suitable piping to distributors mounted on the footplate, usually on the back head of the boiler. The distributors consist of sight-feed glasses fitted with suitable controls, from which the oil is led to the valves, pistons, and also to the driving boxes.

The three-cylinder fast freight engines of the North Eastern are next described and in this connection data are given of some trials made with a 0–8–0 type engine.

The Caledonian locomotive is one of the most powerful 4–6–0 type built for express passenger service. The engine has three cylinders which are all in line and drive on separate axles. The center cylinder drives the leading wheels through a crank axle, and those outside the center coupled wheels through connecting rods 11 ft. long as against 6 ft. 6 in. long for the inside rod.

Several other types are described, among them being a three-cylinder type built for the Great Northern. Here all three cylinders drive on to the center axle and a peculiar valve gear is employed, the valves being of the piston type. There are three of them with inside admission which are operated by two sets of Walschaerts gear applied to the outside motion. (First of a series. *Railway Mechanical Engineer*, vol. 96, no. 12, Dec., 1922, pp. 677–682, 7 figs., *d*)

Effect on Ton-Mile Cost of Reducing Train Loads. Data of tests carried out for the U. S. Railroad Administration on the Illinois Central, from which it would appear that the greatest economy is obtained with full-tonnage freight trains.

The original article describes the method employed in carrying out the tests and gives in tabular form some of the results obtained. Because of lack of space only some of the conclusions can be reported here, as follows:

1 There is no general agreement as to the percentage of tonnage rating which will bring the lowest cost. In the majority of cases this is effected by a load of 100 per cent, but in others the most economical load (considering the cost in one direction only) is about 85 per cent. Each division is governed by its own operating characteristics and no general law appears in the results obtained in this study.

2 The cost per ton-mile decreases as the gross carload increases, due probably to the lower train resistance per gross ton.

3 The cost per ton-mile increased with the delay on the road.

The practical application to that road of the facts developed by the tests is summarized by the following general conclusions in the report of the Federal manager of the Illinois Central:

1 It is not practicable to reduce the train load to avoid overtime because of the increased cost incident to the operation of the necessary additional trains in the direction of heavy traffic to handle the same tonnage and in the direction of light traffic to balance power.

2 To a large extent the cost of handling the most economical train load includes considerable overtime.

3 Increased cost resulting from overtime, like any other wage increase, must be met by increasing facilities instead of by reducing train load. This reduction, in a good many districts, would add train units in excess of present capacity. (*Railway Age*, vol. 73, no. 25, Dec. 16, 1922, pp. 1145–1148, *ep*)

Zeitler Self-Contained Gasoline Motor Truck for Railway-Car Bodies. In this car the truck conforms to a standard railway truck, the frames being cast steel and the engine crankcase forming the hollow bolster. The transmission is hung directly beneath the engine and forms almost an integral part of it. The engine is a four- or six-cylinder horizontal or forced type. The pistons are of the truck types carried in sleeves that act as crossheads and take the wear off the cylinder walls.

The control system is of the electropneumatic type. A small master controller located in the engineer's cab controls the electric starting motor and the carburetor, this latter being governed by a variable-speed electric motor operating the governor attached to the butterfly valve. An electropneumatic arrangement operated also the gear shift.

An interesting feature of the design is the support of the crankcase bolster. This crankcase bolster is spring-supported, a bracket at the ends being extended to a suspension link to permit lateral motion. This suspension link is attached to the frame and at a point on the frame which results in low stresses at the center of the side frame. The springs rest directly on the journal boxes with the ends attached to the side frames, thus giving the crankcase bolster both lateral and vertical movement. The center tie bars between the side frames are bolted in place and therefore can be removed readily. Then by disconnecting the suspension links at the frames, the drive shafts and control connections, the engine and transmission may be removed from the truck for overhauling. (*Railway Review*, vol. 71, no. 25, Dec. 16, 1922, pp. 852–854, 2 figs., *d*)

REFRIGERATION (See also Thermodynamics)

Tests on Ammonia Compressor and Tubular Condensers

Performance of Single-Acting Ammonia Compressor and Tubular Condensers, Geo. A. Horne, Mem. Am.Soc.M.E. Data of tests made in the Tenth Ave. plant of the Merchants Refrigerating Co., New York City. The machine on which the tests were made is a vertical, three-cylinder, single-acting, simple compressor of the enclosed type with pistons 18 in. in diameter and a 20-in. stroke. The machine is direct-connected to a synchronous motor and is operated at 164 r.p.m. The condensers are shell and tube, open type, with water pumped over the top and flowing by gravity through the tubes into an open pan. The condensed liquid is cooled in a double-pipe liquid cooler, from which it passes to a shell and tube brine cooler.

The article describes the equipment used in these tests which appears to be unusually complete for a commercial test. Ammonia was measured by a carefully calibrated meter, which is superior to any method of simply weighing or gaging the liquid during the test.

In view of the fact that the liquid is frequently cooled below the temperature corresponding to the boiling point at the condenser pressure, an interesting chart (Fig. 6) has been arranged so that the percentage of indicated horsepower per ton saved by liquid intercooling may be read directly without detailed calculation.

The chart is subdivided into three parts: (1) the upper left-hand corner gives the relation between condenser pressure, liquid cooling range and B.t.u. removed by water per pound of liquid ammonia;

(2) the lower right-hand corner gives the relation between condenser pressure, suction pressure, and B.t.u. available per pound of ammonia with no liquid cooling; (3) the upper right-hand corner gives the relation between available B.t.u. per lb. of ammonia with no liquid cooling, B.t.u. removed by water per pound of liquid ammonia and percentage of saving in horsepower effected by the liquid forecooling.

The following examples will show how the chart is used:

1 Given: Condenser pressure 190 lb. per sq. in. abs.; suction pressure 20 lb. per sq. in., abs. Find: Available B.t.u. per lb. of ammonia, without any liquid cooling.

Use the lower right-hand section of chart. Enter the right-hand vertical scale at 190 lb. (see dotted line on chart), move horizontally to the right to the 20-lb. suction line; then move vertically down and read 458 B.t.u. on the lower horizontal scale.

2 Given: Condenser pressure 190 lb. per sq. in., abs.; liquid forecooled 20 deg. fahr. (i.e., from 93.1 to 73.1 deg. fahr.) Find: B.t.u. removed by water per lb. of ammonia.

Use the upper left-hand section of chart. Enter the horizontal scale at 190 lb., move vertically up to the 20-deg. line, then move horizontally to the left and read 23 B.t.u. on the vertical scale.

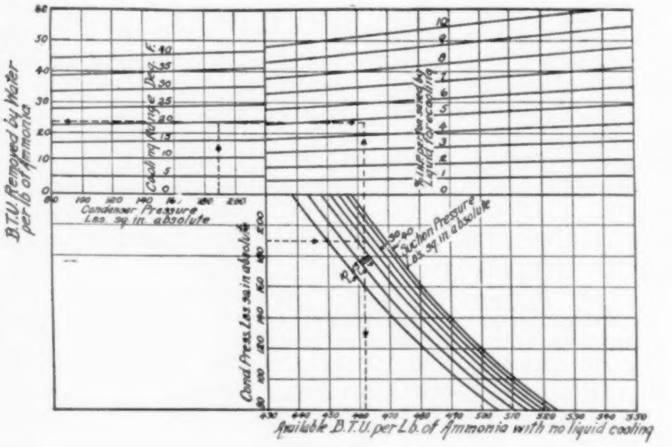

Fig. 6 Theoretical Compression of Ammonia (Simple Compression with Liquid Forecooling)

3 Given: Condenser pressure 190 lb. per sq. in., abs., suction pressure 20 lb. per sq. in., abs.; liquid forecooling range 20 deg. fahr. Find: The i.hp. per ton.

In the lower right-hand section of chart enter the vertical scale at 190 lb.; move horizontally to the right to the 20-lb. suction line, move vertically up; hold this vertical line. In the upper left-hand section of the chart enter the horizontal scale at 190 lb.; move vertically up to the 20-deg. line and then move horizontally to the right until the vertical line previously obtained from the other section of the chart is intersected. At the intersection read 4.8 per cent saving effected by liquid forecooling. From the i.hp. chart we find that the i.hp. per ton for 190 lb. condenser pressure and 20 lb. suction pressure but without liquid cooling is 1.525. Then the i.hp. per ton for the same conditions but with liquid cooling is 1.525 (1 - 0.048) or 1.525 × 0.952 = 1.452.

It should also be mentioned that the theoretical i.hp. as calculated does not include any saving due to water-jacketed cylinders. The machine which was used in these tests was provided with water jackets, which naturally increases the volumetric efficiency and decreases the horsepower per ton. The effect of the jacket water may be calculated from the data as shown for the various tests.

The table in the original article gives a summary of the condenser performance in these tests, of interest because of the high rate of heat transfer obtained with these condensers. (*Refrigerating Engineering*, vol. 9, no. 5, Nov., 1922, pp. 143–151 and 161, 11 figs., *eA*)

SPECIAL PROCESSES

Mauclere System of Pneumatic Handling of Inflammable Liquids. The apparatus is designed primarily for handling kerosene and other inflammable or toxic liquids. It may be used for any other kind of liquids as well, including viscous liquids and those holding solid matter in suspension.

The problem which the designer undertook to solve may be stated as follows: To assure in every instance by means of "interconnecting gage vessels" functioning alternatively the manipulation of the liquids contained in closed reservoirs in any position or dimension and under atmospheric pressure, with the further restriction that these liquids during the period of storing them in the tanks, withdrawing them from the tanks, and intermediary periods, should remain at all times out of contact with the air in order to avoid diffusion of inflammable or toxic vapors.

The original article describes the apparatus in detail as well as test data in connection with the separation. The apparatus is of such a character that it may be applied either to a large stationary plant or to a delivery truck. (*Le Génie Civil*, vol. 81, no. 23, Dec. 2, 1922, pp. 512–516, 9 figs., *d*)

An Improved Tube Rolling Mill

Improved Tube Rolling Mill. Abstract of British patent granted to R. E. Brock, an American. An interesting modification to the construction of tube rolling mills of the continuous type is described under this patent, the object being to secure more efficient operation in the manufacture of seamless tubing. In machines of this nature it is usual to pass the blank through a series of rolls in alignment with a gradually decreasing pass diameter, consecutive sets of rolls being arranged with the planes of maximum pressure perpendicular to one another. This, of course, results in a slight reduction in the diameter of the tube at each pass, with a corresponding elongation of the blank, so that the finished tube leaves the mill at a speed considerably greater than the speed at which the blank enters the opposite end. Actually, under average conditions the increase in speed is found to be approximately

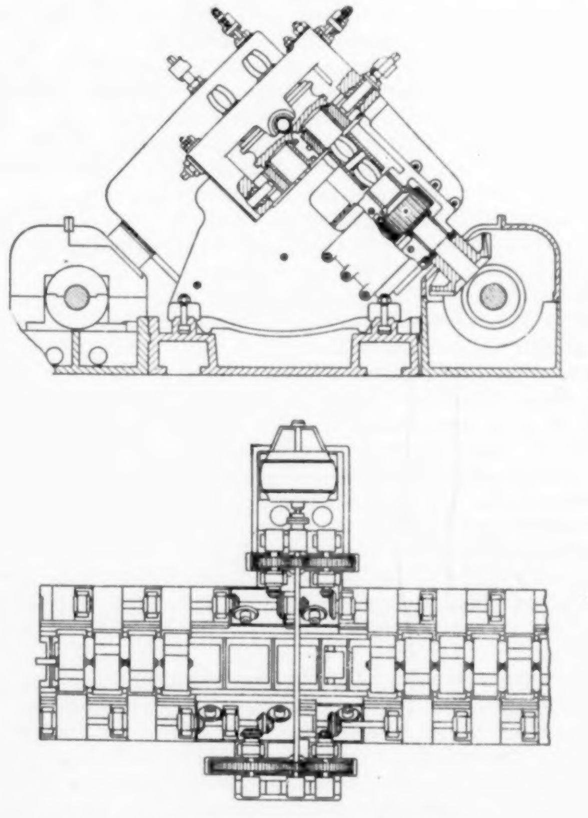

Fig. 7 An Improved Tube Rolling Mill

4 per cent at each pass, and it is to make allowance for this factor that the present improvement has been introduced.

Briefly, it consists in a particularly neat arrangement of the drive to each set of rolls, whereby a progressive increase of the speeds of the respective sets is obtained regardless of the speed of the primary driving shaft. The method of effecting this will be apparent from Fig. 7, which shows a part section through the mill in the

plane of the rolls, and a plan view from which the central sets of rolls has been omitted to show the drive.

It may be seen that the power is taken from a motor direct-coupled to a cross-shaft, which is fitted at either end with a pinion. Each of these pinions engages two spur gears, one on either side, while the spur gears are made with diameters progressively increasing, so that a corresponding range of speeds is obtained.

It may further be stated that the spur gears are keyed to short cross-shafts, each of which carries a bevel pinion at the inner end, and by this means the power is transmitted to four independent longitudinal shafts, which are thus driven at different speeds. The drive is then transmitted to each set of rolls by means of further bevel gears, arranged as shown in the upper view, and these members are also made so that the diameter increases progressively throughout the series. Thus, since alternate sets of rolls are driven from each longitudinal shaft, each pair may be driven at a speed approximately 4 per cent higher than the preceding set by suitably arranging the number of teeth in each bevel wheel.

In other respects the constructional features follow the usual practice with regard to rolling mills. (British Patent no. 185,543, abstracted through *Engineering Production*, vol. 5, no. 112, Nov. 23, 1922, pp. 501–502, 1 fig., *d*)

SPECIAL PROCESSES

Removal of Solid and Liquid Particles from Gases, A. F. Nesbit. Description of an apparatus of a type combining electrical and centrifugal action and intended chiefly for cleaning combustible gases such as coke-oven and blast-furnace gases to be used as a source of heat or motor fuel, though other gases may be similarly handled.

The apparatus may be purely mechanical or combined with certain electrical features. The purely mechanical cleaners do not remove as completely the dusts, tar, and other fogs constituting the recoverable contents of the gases as may be done by the use of combined electrical and mechanical cleaners, but they are free from the disadvantages always present when using electrical systems and they yield such a high percentage of recovery as to suggest a successful stage-refrigerating-type apparatus. (*Blast Furnace and Steel Plant*, vol. 10, no. 12, Dec., 1922, pp. 637–641, 5 figs., *d*)

TESTING AND MEASUREMENTS (See also Thermodynamics)

The Davon Microtelescope and Supermicroscope, F. Davidson. Description of two instruments invented by the author. As regards the microtelescope the principle involved is the application of a short-focus telescope objective in a tube with a series of diaphragms to the microscope, the microscope itself acting as a compound eyepiece. With that attachment, which contained a 6-in. telescope objective inserted into the Abbé rim of the microscope, there was an image of a distant object projected to the plane of the stage; the air image was magnified by the microscope proper and as a result objects could be seen as near as 6 ft. Its range was practically from 6 ft. to infinity. One could then, for example, view the scales of pyrometers or thermometers at any distance, inaccessible parts of machinery, and certain parts of a mine, provided, of course, that the light could be projected on to the object.

One of the characteristics of the observations with this instrument was depth of focus. One could look at an ordinary photograph with the instrument and see it in apparent stereoscopic relief. As an illustration of what can be done with the microtelescope, the author cites the case of a bird's nest which was photographed with this instrument at a distance of 50 yd.

Next, a short-focus attachment was provided for working with objects which, because of their size or shape, could not be put upon the stage of a microscope, for example, minerals, metal fractures, etc. Here a variation of magnification of from 30 to 90 diameters was possible without altering anything but the distance from the object, and even at the higher magnification the relative depth of focus was always maintained.

A third attachment was then provided, namely, another tube in which a microscope objective was placed at one end. At the other end of the tube was another achromatic combination, which the inventor called a collector. This gave the supermicroscope. By means of the collector the image of a microscopic object was projected at some distance beyond it. The position of the air image which was projected depended upon the distance that the collector happened to be from the front of the primary objective. This enabled them to carry the system of direct observation which was begun with a short-focus attachment up to very nearly 1000 diameters and still have sufficient working distance to eliminate an object from above.

It is of interest to note that anything that could be seen with either piece of apparatus could be photographed by it by simply removing the body tube of the microscope and substituting a camera. The same lens was used for telephotography as for observation. With a short attachment it was possible to photograph metal fractures and similar things, and with the supermicroscope the author photographed objects from life size to diatoms under a magnification of 3000 diameters. (*Chemical News*, vol. 125, no. 3270, Dec. 15, 1922, pp. 353–355, see also *Bulletin* of the Institution of Mining and Metallurgy, no. 218, Nov., 1922, pp. 1–5, *d*)

THERMODYNAMICS

Heat Transmission—Heat Measurements—Room Thermometers

New Methods and Investigations for Determining Heat Transmission, Prof. Oscar Knoblauch. Discussion of the meaning of the various coefficients of heat transmission and methods for their determination, including methods of measuring the temperature of the various elements entering into the problem.

The author devotes particular attention to the determination of the coefficient of heat transmission α which is the heat lost per unit area, say, 1 sq. m., of a surface, such as the wall of a building, to the ambient medium, such as air, per unit of time per hour per degree (centigrade) temperature difference. The great difficulty in determining this coefficient lies in the exact measurement of the air and wall temperatures.

In particular, as regards the measurement of air temperature, this should be measured in fairly close proximity to the wall. The difficulty of doing this, however, lies in the fact that if the thermometer is placed close to the wall, its reading may be materially affected by radiation losses, the direction of which would depend on whether the wall is colder or warmer than the thermometer. In the former case the thermometer would be losing heat by radiation to the wall and would show less than the true air temperature. In the latter case it would gain heat from the wall and would show an air temperature higher than the wall.

Experiments recently carried out by H. Hausen in Germany show that this error in thermometer reading may be quite material. All attempts to obviate this by such means as enclosing the thermometer in a metal sheath or using a rapidly moving thermometer or blowing the air over the thermometer and the like, are unsuitable for this particular purpose because they materially modify the conditions of the experiment.

Theoretically, the error in measurement due to the error of absorbed radiated heat may be considerably reduced by providing a thermometer with a surface of proper characteristics, namely, one that will absorb as little light and heat radiation as possible. The glass tube usual in mercury thermometers allows the majority of light rays to pass through, which therefore reach the mercury and are reflected by it. On the other hand, however, the glass absorbs powerfully the shorter-wave heat radiation and thereby contributes to the error in measurement discussed above. Since, however, gold and silver absorb only 2 per cent and nickel 5 per cent of the heat radiation, a substantial reduction in error due to heat absorption may be brought about by gilding, silvering, or nickeling the thermometer tube. If gold and silver are selected the error in measurement falls to about 3 per cent of the error of a thermometer with a plain glass tube, and this may be safely neglected in the majority of practical applications.

For measurements of precision, however, the readings of a thermometer even with the gilt and silver tube, are not sufficiently exact. There is, however, a way of obtaining an extremely exact reading by using two thermometers having tubes of different heat-absorption capacity, for example, one of plain glass and the other

gilt. In this case the true temperature of air t_0 may be expressed as—

$$t_0 = t' - K\,(t'' - t')$$

where t' is the reading of the thermometer with the gilt tube; K a coefficient depending on the constants of the two thermometers (in this case 0.15); and t'' the reading of thermometer with the higher heat absorption, as, for example, the plain glass tube.

If, for example, with such a double-thermometer outfit the glass-tube thermometer reads 23.0 deg. cent. and the gilt tube 20.0 deg., then the true temperature of the air t according to the above equation is—

$$t = 20 - 0.15\,(23 - 20) = 19.55 \text{ deg. cent.}$$

If such a double-thermometer outfit is suspended near a *cold* wall the temperature of the gilt-tube thermometer will be higher than that of the glass-tube thermometer and the temperature of both will be lower than the air temperature, all this being due to the loss of heat from the thermometers to the wall by radiation.

From this the author proceeds to the discussion of the ordinary method of measuring room temperature by means of a thermometer suspended on the wall. He finds that with the usual construction such a thermometer reads the room temperature fairly closely only where the room is so heated that it is maintained at a fairly constant temperature for a length of time sufficient for a fairly close equalization between air and wall temperature to be obtained. However, where the room is not heated for a greater part of the day and is only supplied with considerable amounts of heat for comparatively short intervals, so that the walls are not fairly cold, the temperature readings of a thermometer hung on a wall are extremely misleading. This is particularly so as the mercury bulb is enclosed in a wooden frame, and in addition is covered by perforated metal. Free contact with air is therefore materially precluded, while heat transfer between the wall and thermometer by conduction and radiation are encouraged. Such a thermometer is more suitable for measuring the wall temperature than the air temperature.

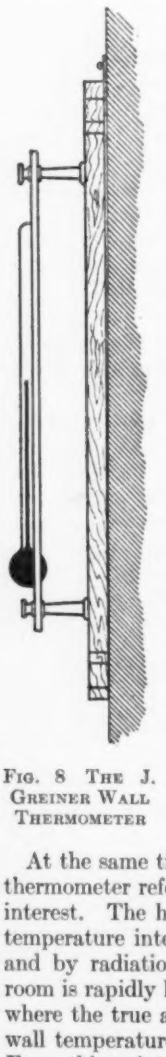

Fig. 8 The J. Greiner Wall Thermometer

A German concern is building a wall thermometer of the type shown in Fig. 8. Here the thermometer is supported on a wooden base suspended on the wall in the usual manner, but it is placed not in the base itself but on a milk-glass scale about 2 cm. (0.8 in.) from the wooden base. The mercury bulb is therefore freely in contact with the air while the wooden base protects the thermometer against a large share of heat radiation interchanged between it and the wall.

At the same time it is a question which of the two types of wall thermometer referred to above gives indications of greater practical interest. The human body in a room is subject to two kinds of temperature interchanges: namely, between the body and the air, and by radiation between the body and the walls. Where the room is rapidly heated up there may be a feeling of chill in a room where the true air temperature is as high as 68 deg., provided the wall temperature is low, and the latter may be as low as 50 deg. From this point of view a thermometer exposed to radiation losses to the wall loses more than one that is not so exposed, and, for example, under the above air and wall conditions a thermometer in a room having a true air temperature of 20 deg. cent. (68 deg. fahr.) will read only 17.3 deg. cent. (63 deg. fahr.).

The next problem which the author discusses is the measurement of the true wall temperature, which means the temperature of the surface of the wall exposed to the air. Here thermocouples present a sufficiently reliable means.

The author proceeds next to the discussion of methods for determining the heat transmission through walls of various tubes, hollow or solid, by means of numerical calculations, and shows that these have been developed to such an extent as to give results sufficiently reliable for all practical purposes. He points out also that of late a good deal more attention has been paid than formerly to the ability of walls to resist heat losses because of the greater cost and scarcity of fuel.

An extensive discussion of heat losses through windows concludes the article. In this case the author considers losses due both to heat flow through the window materials and air leakage due to lack of tightness in the window joints. In this connection extensive experiments made in Germany are quoted. (Paper before the annual meeting of the German Refrigerating Society, Munich, July 18, 1922, abstracted through *Zeitschrift für die Gesamte Kaelte-Industrie*, vol. 29, no. 10, Oct., 1922, pp. 177–183, 7 figs., *eg*)

VARIA

Industrial Situation in Japan. Data taken from a report by Sir E. T. F. Crowe and G. B. Sansom of Tokyo, issued by the British Department of Overseas Trade and dealing with the commercial, industrial, and financial situation in Japan during 1921 and up to June 30, 1922.

From this report it would appear that contrary to the common impression Japan is a country of high production costs. This is illustrated in the case of copper. Japan is one of the largest producers of copper in the world. Before the war her output averaged 60,000 tons a year, and next to silk and cotton yarns copper was the most important export. The principal mines were owned by a few of the richest families of the country and the copper business was looked upon as being very prosperous and profitable. In 1919 and 1920 for the first time copper began to be imported as consumption increased, while production fell off owing to increased costs. The imports fell off in 1921 but increased again in 1922. The copper-mine owners declared that they were unable to produce copper at a price which could compete with the imports from America, and the government, in order to protect an important domestic industry, accordingly increased the duty from 0.20 yen per 100 kin to 7 yen. For a short period imports ceased, but last June they were again beginning to come in.

It is estimated that the consumption of copper in Japan is about 6000 tons a month, while production has dropped below 5000 tons. In 1917 Japan produced on an average 10,000 tons monthly. The problem facing the mine owners is whether to attempt to produce more economically and meet competition or to maintain the price at a high level by reducing output.

The latter uneconomic policy has been followed by the colliery owners who curtailed output by $17^1/_2$ per cent in order to bolster up prices. It is this tendency, the report points out, of continually forcing up prices and preventing them from finding their natural level which must seriously damage Japanese trade unless it is remedied in time. The immediate effect has naturally been to keep the cost of living at the high level to which it mounted during the war.

As regards the iron and steel industries, it was pointed out in another report that Japan has made rapid strides in its wire-manufacturing industry and today is turning out wire rope, plain wire, and wire nails of good quality. The production of these commodities, however, is not sufficient to meet the needs of the Japanese, resulting in heavy purchases of foreign wire goods. The demand for wire products is principally in galvanized plain and bright wire and wire nails. To a less extent there is a call for wire rope and cable, insulated wire and cable, woven wire fencing, wire cloth and screening.

Boiler plates for the Japanese-made locomotive, war vessel, steamship, and other equipment are largely imported. In May, 1922, 1839 tons of boiler plates and 2189 tons of other steel plates of American manufacture were taken by the Japanese. (*The Electrical Review*, vol. 91, no. 2344, Oct. 27, 1922, pp. 605–606, *g*)

CLASSIFICATON OF ARTICLES

Articles appearing in the Survey are classified as *c* comparative; *d* descriptive; *e* experimental; *g* general, *h* historical; *m* mathematical; *p* practical; *s* statistical; *t* theoretical. Articles of especial merit are rated *A* by the reviewer. Opinions expressed are those of the reviewer, not of the Society.

Progress in the Art of Power Development

By A. G. CHRISTIE,[1] BALTIMORE, MD.

THE Power Division concerns itself with all problems involved in the generation and transmission of power. Among the more important problems are those concerning the development of underlying theory, the design, construction, and operation of all classes of power machinery, both steam and hydraulic, the transmission and distribution of power, and economic questions concerning power development.

Since the greater portion of our power supply is developed from steam at the present time, much attention has been devoted to factors concerning the design and construction of steam stations. The increased costs of fuel and of labor have had a very material affect on plant design. Emphasis is now placed on the operation of the plant as a whole rather than on prime mover or boiler alone. This has led to careful study of the heat balance of the complete plant. Such studies often develop possibilities for improvements and have a very stimulating effect on the management. In new designs much thought is devoted to heat-balance studies of various schemes before a final selection is made. Minor heat losses such as those in the generator cooling air, transformer losses, bearing losses, and gland leakage are now being recovered and utilized. Systems of operating auxilaries to secure the lowest B.t.u. per kw-hr. of net station output are being developed which differ radically from former practice. In some cases house turbines are employed with motors on all auxiliaries. These, however, will probably be omitted from future stations where economizers are not used and all power for auxiliaries will be furnished from the main turbines which will be of the multiple bleeder type. Certain of the essential auxiliaries will be equipped with synchronous motors and a duplex steam drive in reserve that will pick up load in an emergency and will operate certain other auxiliaries with the synchronous motor acting as a generator.

Substantial progress is being made both in size and in designs of large turbo-generators. Types built under the stress of war conditions developed certain weaknesses, and as a result all the large companies both here and abroad have been redesigning and improving their machines. The newer designs are more compact and better built. The ideal turbine will provide the steam with the smoothest and most direct expanding passages free from sudden changes in cross-section or in direction, and with stationary and revolving guides or buckets whose curved surfaces offer the minimum of shock to the high-velocity steam. This ideal is being approached in recent designs. A uniform distribution of steam in the exhaust pipe and the utilization of the velocity energy leaving the blades are also attempted in some of the new units.

Multi-stage bleeding is becoming standard practice with large steam turbines. Engineers will soon demand that the builders themselves provide the turbine and bleeders and base all guarantees on the performance of the unit as a whole, resulting in better design and improved thermodynamic performance.

The characteristics and limitations of small steam turbines are becoming better known. New designs embody substantial improvement over earlier types. There is still need for further improvement in these units intended for auxiliary service. Small steam turbines have not been able to compete on the basis of economy with uniflow engines.

There is a decided tendency at the present time to increase steam pressures and temperatures. Mr. Orrok has ably discussed this subject in his recent paper on The Commercial Economy of High Pressure and High Superheat in the Central Station. He concludes that 1200 to 1500 lb. steam pressure is commercial but that temperatures are limited to about 750 deg. fahr. until better materials can be developed for valves and piping. The Benson system now being considered in Great Britain goes to pressures in the neighborhood of 3200 lb. per sq. in., but while sound theoretically has not been commercialized. Recent announcement of plants to use 550 lb. pressure seem to indicate conservative design as far as ultimate limitations are concerned.

Mr. Orrok pointed out that most of our discussions on high pressures and temperatures are academic in view of the lack of definite knowledge of the properties of steam under these conditions. The Steam Table Research when completed will be of inestimable value to engineers. Much discussion of the so-called "supersaturation" state of steam has developed in England. American engineers desire to be further enlightened on this subject, which up to the present time has not received much attention in this country.

The steam boiler and it auxiliaries are prime essentials in power generation. Boilers are steadily increasing in size and height. The Power Division has directed its attention to a study of heat absorption in boilers, and this is progressing satisfactorily under the direction of Mr. E. B. Powell. It is hoped that an early report can be made to the Division. The introduction of powdered fuel with increasing furnace temperatures has made the employment of water screens necessary and has suggested the use of radiant-energy superheaters in the side walls. The influence on furnace temperature of these added heat-absorbing surfaces will be watched with great interest, and will have considerable effect on future furnace design.

Boilers are now operated for varying periods at high ratings. This can only be done where the tube surfaces are free from scale and the feedwater free from foaming alkalis. Hence the feedwater must be kept pure. Many stations now employ evaporated make-up. If low-priced evaporators were available in small sizes it would be advisable to furnish distilled make-up for feedwater in a great many of our small plants where only impregnated water is available.

Rising coal prices have resulted in the wider adoption of economizers, which in high-pressure plants have steel tubes. Several new problems have been presented to engineers by the adoption of these steel tubes. In the first place corrosion is very rapid and very destructive if any oxygen remains in solution in the feedwater. Hence various methods and equipment have been devised to deaërate the feedwater. Difficulty with corrosion of the outside of economizer tubes has been encountered where coals high in sulphur are used. A study of the dewpoints of sulphur compounds may reveal the cause of this difficulty.

In plant economy the B.t.u. per kw-hr. developed has been steadily reduced and will be further lowered when some of the newer ideas on heat balance have been fully developed.

There is a wide interest nationally in the development of water power, and many new undertakings are being planned. Engineers have devoted their attention to the development of turbines of increasing size for large power developments and to new designs to operate efficiently under low heads. Definite progress has been made in improving the overall efficiency of turbines by the use of the hydraucone, the improved draft tube, and the regainer. Several of these developments have been discussed at meetings under the auspices of the Power Division and papers on new developments are planned for coming meetings.

Reduction gears produced during the urgency of the war period were in many cases not of the best materials nor carefully made. Gears of recent manufacture have been greatly improved and are now regarded as standard equipment for many purposes.

The increased interest being given to the question of conservation of the valuable products of coal and to the low-temperature carbonization process should make power engineers give some thought to the ultimate plant of the future. A scheme is already under consideration for at least one large station where a low-temperature carbonization process adjoining the steam plant will furnish rich gas to the city mains for domestic purposes and a low-volatile coke that burns without smoke for heating purposes. This semi-coke in pulverized form will serve as the fuel supply to the boiler furnaces of the electric plant. The motor fuels, alcohol, creosote, fuel oils, ammonium sulphate, and other valuable by-products will be available for sale. In the near future the power engineer may have to be a gas-house man as well as a steam-station expert if the present tendency continues.

[1] Professor of mechanical engineering, Johns Hopkins University. Retiring Chairman Power Division, A.S.M.E.

ENGINEERING RESEARCH

A Department Conducted by the Research Committee of the A.S.M.E.

Personnel Methods—Recognition and Classification of Human Abilities

DURING the last decade there has grown up in industry a new branch of technology that has received the appropriate descriptive name of "human engineering." The development has come about through an increasing demand for means to assist managers in selecting men better suited, from the points of view of employee and of employer, to the work they have to do. There is wide demand for reliable information about the methods which have appeared in the form of mental tests, trade tests, rating scales, and other means of discovering human qualifications.

Standardized tests and measurements are widely sought by engineers, but the hazards incident to the activities of charlatans prevent the full utilization of this contribution of modern science. Recognizing the importance of the movement and the need for careful scientific appraisal of available methods, the National Research Council has initiated various activities for the benefit of managers who are dealing with human problems. To this end the Research Information Service of the Council is giving attention to the compilation of reliable information about personnel matters.

As a free clearing house for the promotion of research and its applications in industry, the Research Information Service is prepared to furnish critical information about modern personnel methods. Among the mechanisms especially designed for this informational service there is maintained a personnel file in which consulting psychologists, personnel specialists, and other reputable experts on problems of human adjustment are listed. Files of information about available tests and their particular usefulness are also maintained.

With specialists on these problems in the Council offices and with files of information at their disposal, mechanical engineers may profitably appeal to the Research Information Service for facts about methods of personnel classification. All communications should be addresses to Research Information Service, National Research Council, Washington, D. C.

Research Résumé of the Month

A—Research Results

Explosives and Explosions A1-23. Study of Explosions of Gaseous Mixtures. The immediate object of the series of tests reported in the most recent Bulletin (No. 133) of the Engineering Experiment Station of the University of Illinois was a study of the physical phenomena involved in the explosions of various mixtures of illuminating gas and air. Prof. A. P. Kratz and Mr. C. Z. Rosecrans are the authors. The bulletin includes the determination of the effect of different-shaped explosion vessels, and of turbulence at the time of ignition, as well as a study of the heat loss from the burning mass of gas. Any problems connected with the chemical changes involved have not been considered within the scope of the investigation.

An investigation of explosions of gaseous mixtures at constant volume was undertaken by the Engineering Experiment Station in 1914, when C. R. Richards outlined an extensive series of experiments and defined the scope of the investigation which was to follow. Apparatus was designed and a few results were obtained showing the effect of the shape of the explosion vessel and of turbulence at the time of firing, but it was found necessary to discontinue the work. In 1919 part of the apparatus was redesigned and work was resumed. This bulletin is a preliminary report of the methods pursued and the results thus far obtained.

Address the authors in care of the University of Illinois, Engineering Experiment Station, Urbana, Illinois.

Gases, A1-23. Study of Explosions of Gaseous Mixtures. See *Explosives and Explosions A1-23.*

B—Research in Progress

Ceramics and Glass B1-23. Influence of Burning Fuel Oil on Refractories. See *Refractories B1-23.*

Fluid Flow B1-23. Flow of Water in Vertical Pipes. A considerable amount of laboratory work has been completed in an investigation of the possible rates of flow through vertical pipes. This study is being conducted at the University of Illinois by Mr. W. J. Putnam and is to include work on pipe of various lengths and with different entrance conditions. The pipes discharge into the air in all cases and the tests, therefore, apply to flow through down spouts, drains, industrial piping systems, etc.

Framed Structures B1-23. Stresses in Statically Intermінate Frames. Bulletin No. 108 of the University of Illinois issued in 1918 presents equations for moments in frames due to any type of load. The work which Mr. W. M. Wilson has completed during the past year on this subject consists of the derivation of the corresponding equations for the stresses due to particular loads. This material will be published shortly as one section of the Structural Engineer's Handbook Library.

Framed Structures B2-23. Web Stresses in Beams. A study of the effect of spacing of stirrups and a comparison between the actions of rectangular and T-beams is now being made under the direction of Prof. A. N. Talbot at the University of Illinois. During the past year tests on 20 large rectangular and T-beams have been completed.

Gas, B1-23. Efficiency in Gas Combustion. An investigation has been recently undertaken by Mr. F. E. Vandaveer at the Engineering Experiment Station of the University of Illinois to determine the possible extensions in the use of fuel gas in the industries. As this study progresses it is proposed to take up (*a*) the catalytic chamber, (*b*) thorough mixing of air and gas, and (*c*) the effect of temperature at time of combustion.

Gases, B2-23. Amount of Water in a Nearly Dry Gas. An electrical device for detecting and approximately indicating the amount of water in a nearly dry gas, upon which a considerable amount of work was done several years ago at the Bureau of Standards, has been again tried out with surprisingly good results. The detector appears to have lost none of its sensitivity during nearly three years in the laboratory. This would indicate that it may be relied upon as an alarm device in the liquefaction and nitrogen-fixation industries where such a device is greatly needed. The Bureau hopes to prepare and publish a description of the device in the near future.

Mechanics, B1-23. Web Stresses in Beams. See *Framed Structures B2-23.*

Mechanics, B2-23. Fatigue of Metals. Bulletin No. 124 of the University of Illinois describing this important investigation by Professors Moore and Kommers was mentioned in these columns in the February, 1922, issue. It is now possible to announce that a second bulletin by the same authors is in preparation which will describe new tests of materials under a combination of steady tension and reverse bending and also under reversals of direct axial stress.

Non-Ferrous Metals B1-23. Contraction and Shrinkage of Non-Ferrous Alloys as Related to Casting Practice. One of the fundamental problems under investigation by the Bureau of Mines in its alloy work is the contraction of a series of light aluminum alloys, particulars of which will soon be published as Bulletin 287 of the Bureau. An investigation is now under way on the contraction of a series of commercial brasses and bronzes.

While the contraction of alloys is only one of the factors that bear on cracks in castings, as well as on the casting qualities of any alloy, still it is important; and comparative figures as to the contraction of various alloys will serve as a guide to their casting qualities and cracking tendency.

A "Report of Investigation" Serial No. 2410 prepared by Mr. Robert J. Anderson is a short discussion of the technical aspects of contraction in relation to foundry practice, and has been published in response to numerous inquiries for information, with the recommendation of the Advisory Committee.

On the basis of the available information, the following conclusions may be drawn as to the contraction and shrinkage of alloys as related to foundry practice:

1 The linear contraction of any alloy is a function of the exact chemical composition of that alloy, and relatively small amounts of impurities affect the contraction.

2 The wide variation in contraction among the alloys of a given class—for example, brasses, bronzes, or aluminum alloys—indicates that it is poor practice to employ a rough figure as the contraction of alloys of a given class in general, since by so doing much difficulty arises in producing master patterns and in obtaining castings with a minimum of wasters.

3 Theoretically, the higher the pouring temperature the greater the contraction, as less metal can be held in a mold cavity at a higher temperature than at a lower temperature. Actually, pouring at an intermediate temperature gives greater contraction than pouring at a high temperature or a low temperature on account of the effect of gas evolution in the case of alloys poured at high temperatures. The evo-

lution of gas referred to causes actual expansion, thus interfering with normal contraction and yielding less contraction.

4 Other things being equal, the smaller the cross-section, the less the contraction for a given length.

5 Other conditions being the same, the greater the length for a given cross-section the less the contraction.

6 The linear contraction of alloys is a function of the kind of mold employed, the contraction being greater in chill molds than in sand molds, other conditions being the same.

7 In sand molds the contraction in a casting depends upon the contour of the pattern, the mass of the casting, and the size and distribution of the gates and risers—that is, the method of molding.

8 Gas occlusion, owing to a high melting temperature, causes less contraction owing to evolution of the gas when the metal freezes.

9 The extent to which piping occurs on casting an alloy in an open-top ingot mold is a factor in determining the suitability of an alloy for casting purposes, as this is an indication of the contraction in volume. In general, the less the depth of pipe, the less the contraction in volume.

Railroad Rolling Stock and Accessories B1-23. Action of Exhaust Jet in Front End of Steam Locomotives. This investigation has been undertaken at the University of Illinois by Prof. J. M. Snodgrass to determine the action of the exhaust jet in producing draft. The study will seek to determine the influence upon draft of exhaust steam pressure and volume, jet velocity, and form of jet nozzle. It is intended to begin this work by means of a small model so constructed as to permit visual observations of the actions within the front end as well as the usual observations for draft, steam pressure, etc. Any conclusions or generalizations arising from work with the model will be checked on the locomotive in the laboratory, and possibly in road tests.

Railroad Rolling Stock and Accessories B2-23. Tractive Effort of Steam Locomotives. Tests for the accumulation of additional experimental data are now in progress, being conducted jointly by the Illinois Central Railroad and the University of Illinois. The specific object for this investigation is the determination of a method for predicting the tractive effort of locomotives, knowing their dimensions and principal operating conditions. Address Edward C. Schmidt, care of the University.

Refractories B2-22. Study of Electric-Furnace Refractories. As the development of refractories for electric furnaces is now being undertaken, it is desirable to have a method and apparatus for measuring their conductivity at advanced temperatures, states the Federal Bureau of Mines. Moreover, data in regard to the conductivity of existing refractories at temperatures above 1400 deg. cent. are meager. It is proposed to study the leakage factor through refractories. The method of attack has been worked out and the furnace designed, the material for which is arriving at the U. S. Bureau of Mines ceramic experiment station at Columbus, Ohio.

Steel B1-23. Gage Steels. Progress reports on the work so far done by the Gage Steel Committee were considered at the meeting held in New York on November 17th and a program outlined for continuing the work. The samples of a steel selected by the committee for investigation have been received and preliminary work is well under way. Dimensional changes on hardening and changes with time after hardening are being studied; also the rate of wear of different specimens which have been subjected to various heat treatments.

C—Research Problems

Framed Structures C1-23. Strength and Action of Bolted and Riveted Connections. A number of companies interested in this investigation have supplied the University of Illinois with a considerable amount of material with which to begin work. Professors A. N. Talbot and A. F. Moore are to undertake this study as soon as the necessary assistants and machines are available.

F—Bibliographies

Explosions and Explosives F1-23. Gaseous Explosions. This bibliography contains references, not only on the subject of the physical phenomena involved in the explosions of gaseous mixtures but also in regard to the chemical transformations and other allied phenomena. Numerous references of a mathematical nature, particularly in regard to the subject of explosion waves, are also included, as well as some few references describing instruments and apparatus used in investigations of gaseous explosions. It consists of twenty-one closely printed 6 by 9-in. pages and forms Appendix C of the University of Illinois Bulletin No. 133.

Gases F1-23. Gaseous Explosions. See *Explosions and Explosives F1-23.*

Highways F1-23. Projects in Highway Research Current or Recently Completed. The October, 1922, Bulletin of the National Research Council is devoted entirely to a report on the Census which the Advisory Board on Highway Research recently completed. In Part IV, 479 research projects in highway engineering and highway transport are listed and briefly described. These projects are listed under the following five main headings: Economics, Operation, Design, Construction, and Material. Address The National Academy of Sciences Washington, D. C.

Road Materials and Equipment F1-23. Projects in Highway Research Current or Recently Completed. See *Highways F1-23.*

Framed Structures F1-23. Riveted Joints. The literature on this subject both European and American has been very carefully listed and briefly abstracted by Dr. A. H. Stang, connected with the Structural and Engineering Materials Section of the Bureau of Standards. It consists of twenty-six closely typewritten pages. Address Bureau of Standards, Washington, D. C.

Progress Report of A.S.M.E. Research Committee on Fluid Meters

After years of effort on the part of the Research Committee on Fluid Meters of The American Society of Mechanical Engineers, its report is complete and will be issued in pamphlet form within a few weeks.

This report takes the form of a reference book on flow meters of all kinds. It contains not only such practical instruction and information, including formulas, constants, etc., as may be needed by the actual or prospective user, but also more general information —the physical principles of design and operation—which may be useful to students, designing engineers, and inventors.

Part I treats the general types of flow meters as well as the principles and methods involved and gives information which may, in many cases, be applicable to various commercial meters. In this part individual makes of instruments are not discussed in detail, but are referred to only incidentally or for illustrative purposes. The general physical principles are in the body of the text, while the derivation of formulas and the refinements of the theory involved have been placed in the appendices which accompany the report.

In Part II will be found more detailed information concerning the practical use of all the flow meters now in common use. This information has been obtained from both users and makers and includes descriptions of commercial meters, and particulars with respect to operating characteristics, influence of installation, and the testing of meters.

Flow meters are of great and rapidly increasing importance, but hitherto the information available on this group of instruments has been incomplete. The material forming Chapters 1, 2, 3, 4, 5, 6, and Appendix C, were recently rewritten. This material contains a new presentation of the subject based on a mathematical analysis which is more specifically applicable to fluid flow than Bernoulli's theorem. The analysis did not include the additional experimental data now given in the report, but these data required no serious modification of the text.

The most important modern advance in experimental aerodynamics and hydraulics is the application to them of dimensional analysis. This is absolutely indispensable to an understanding of the behavior of moving fluids, for the phenomena of fluid motion are so complicated as to defy analysis by any other known method. The use of this method is especially valuable, in that it makes possible the reconciliation of data obtained from experiments with the venturi tube and the thin-disk orifice which were formerly thought to be irreconcilable. These data are now shown to be mutually confirmatory.

A few of the chapter heads will serve to indicate the scope of the material covered by this report: Classification of Fluid Meters, Weighing Meters, Volumeters, Current Meters, Head Meters including General Principles, Venturi Meters, Flow Nozzle, Thin-Plate Orifice, Pitot Tube, Area Meters, Force Meters, and Thermal Meters.

The Report was prepared and signed by the following men who constitute this special Research Committee on Fluid Meters: Messrs. R. J. S. Pigott, Chairman, J. M. Spitzglass, Secretary, E. G. Bailey, L. L. Borden, G. S. Coffin, A. R. Dodge, L. M. Goldsmith, F. G. Hechler, Horace Judd, Leo Loeb, P. S. Lyon, W. Maplesden, H. N. Packard and C. G. Richardson.

Those desiring copies of this report should address the Secretary, care of The American Society of Mechanical Engineers, 29 West 39th Street, New York, N. Y.

CORRESPONDENCE

CONTRIBUTIONS to the Correspondence Department of MECHANICAL ENGINEERING are solicited. Contributions particularly welcomed are discussions of papers published in this journal, brief articles of current interest to mechanical engineers, or comments from members of The American Society of Mechanical Engineers on activities and policies of the Society in Research and Standardization.

Mathematical Determination of the Modulus of Elasticity

TO THE EDITOR:

The letter in the December issue of MECHANICAL ENGINEERING from Mr. David Guelbaum, entitled Mathematical Determination of the Modulus of Elasticity, is a very interesting deduction from what is commonly known as the general equation of the elastic curve. It should be noted, however, that while a high degree of precision is claimed for the method and that while the derivation of the equations is mathematically correct, Mr. Guelbaum has apparently ignored the fact that he started with a formula which has decided limitations. The general equation of the elastic curve which states that the bending moment at any section of a bent bar is equal to the modulus of elasticity of the material multiplied by the moment of inertia of the section and divided by the radius of curvature at that section ($M = EI/\rho$), depends for its derivation on the assumption that the radius of curvature is practically infinite. This is far from being the case in the use of Mr. Guelbaum's method.

The equations derived and the mechanical device suggested may prove useful for approximate results where no testing machine is available, but I doubt very much the possibility of attaining the degree of accuracy with this device that is attained with the ordinary testing-machine method.

New York, N. Y. WM. R. BRYANS.

The Entropy of Saturated Steam: Its Relation to Specific Volume

TO THE EDITOR:

The object of this communication is to call attention to what appears to be a remarkable relation between the entropy of saturated steam and its specific volume. While the total entropy of the steam from water at 32 deg. fahr. cannot be calculated directly from the specific volume, if this quantity is determined for one temperature and pressure, then the entropy for all other temperatures and pressures can be calculated from the respective volumes, without the use of other data.

If V_0 and s_0 represent respectively the specific volume and the known entropy of saturated steam of any pressure; V_1 and s_1 the specific volume and entropy of steam at some higher pressure; and if V_2 and s_2 the same quantities at some lower pressure; then—

$$s_1 = s_0 - \phi\,(\log V_0 - \log V_1) \qquad [1]$$

$$s_2 = s_0 + \phi\,(\log V_2 - \log V_0) \qquad [2]$$

If the common system of logarithms is used, $\phi = 1/5 = 0.2$.

In the examples which follow, values for V and s are taken from Goodenough's steam tables, 1915 edition. Steam at atmospheric pressure has been selected for the starting point of the calculations, on the assumption that our experimental knowledge of the properties of steam is probably more exact for that pressure than for other pressures. Therefore—

$$V_0 = 26.81 \text{ cu. ft.}$$
$$\log V_0 = 1.4282968$$
$$s_0 = 1.7589$$

For steam at 300 lb. pressure,

$$V_1 = 1.545 \text{ cu. ft.}$$
$$\log V_1 = 0.1889285$$

From Equation [1],

$$s_1 = 1.7589 - 0.2\,(1.4282968 - 0.1889285) = 1.5110$$

The entropy of steam at 300 lb. pressure in Goodenough's tables is 1.5092. The calculated value of 1.5110, while $1/839$ larger than the tabulated value, is smaller than the value assigned by Marks and Davis or by Peabody.

For steam at 1 lb. pressure,

$$V_2 = 333.3 \text{ cu. ft.}$$
$$\log V_2 = 2.5228353$$

From Equation [2],

$$s_2 = 1.7589 + 0.2(2.5228353 - 1.4282968) = 1.9778$$

The value given in Goodenough's tables is 1.9775, which is practically identical.

Table 1 gives the entropies as calculated by Equations [1] and [2] for several intermediate pressures between the extremes of 300 lb. and 1 lb., as well as the corresponding values given in the standard steam tables of Goodenough, Peabody, and Marks and Davis. It will be observed that wherever the calculated values differ appreciably from the tabular values of Goodenough, they lie between the values of Goodenough and those of the other authorities.

TABLE 1 VOLUME AND ENTROPY OF SATURATED STEAM

Absolute pressure, lb.	Volume, cu. ft.	—Total Entropy from 32 Deg. Fahr. According to— Eq. [1] and [2]	Goodenough	Peabody	Marks and Davis
300	1.545	1.5110	1.5092	1.5130	1.5129
200	2.292	1.5453	1.5456	1.5459	1.5456
125	3.593	1.5843	1.5858	1.5840	1.5859
50	8.53	1.6594	1.6601	1.6581	1.6581
14.7	26.81	....	1.7589	1.7566	1.7565
5	73.5	1.8465	1.8456	1.8435	1.8432
1	333.3	1.9778	1.9775	1.9762	1.9754

A study of reversible heat changes in a theoretically perfect gas indicates very emphatically a direct relation between change of volume and change of entropy. For example, if a perfect gas having the characteristics of air at ordinary pressures and temperatures—the constant value of PV being $53.35T$—be expanded from V_a to V_b by any reversible heat process, the initial and final temperatures being the same, the numerical value of the increase of entropy will be equal to—

$$\frac{1}{6^1/_3}\left(\log_{10}\frac{V_b}{V_a}\right)$$

In attempting to establish a similar relation in the case of a saturated vapor it was thought that since the change of volume occurred only during the process of vaporization, the direct relation would exist between the volume and the entropy of *vaporization*. After failing hopelessly to establish this apparently logical relation, it was found, by taking a seemingly forlorn chance, that the total entropy must be considered—i.e., the increase of entropy from the beginning of the liquid stage—until the transformation into a saturated vapor is complete. This indicates that the so-called "heat of the liquid" plays its part in increasing the volume of the fluid when the opportunity presents itself.

It is interesting to note that in the entropy calculations described the factors of heat and temperature are absent. Possibly they may enter indirectly through the volume ratios.

There may be a more exact value for ϕ—Equations [1] and [2]—than 0.2, but inasmuch as it gives results differing from those of recognized authorities less than these authorities differ from each other, its convenience justifies its use.

There are indications that some similar relation may exist between the volume and the entropy of superheated steam, but it is perhaps not so simple and direct. The relation changes very abruptly and very substantially at the instant that the region of superheat is entered.

This volume-entropy relation was not evolved by any scientific

investigation. It is merely something on which the author stumbled accidentally, and which on examination looked interesting. It is passed along in the hope that it may suggest a new line of thought to some one who is skilled and experienced in research work of this general character.

HENRY E. LONGWELL.

Eastwood, N. Y.

STRESSES IN ELECTRIC-RAILWAY MOTOR PINIONS

(*Continued from page 95*)

elastic method had predicted is to be found in the fact that beyond the elastic limit the stress-and-strain relation no longer follows Hooke's law. Therefore the stresses set up in the steel pinions by the shrinking process no longer correspond with those set up in the celluloid model.

While the flat shape of the break in Fig. 9 is one limiting case (torque without radial shrinking pressure), Fig. 8 may be considered as the other limiting case (radial shrinking pressure without torque), showing a V-shaped fracture for which the angle of the V has become equal to zero.

It may be concluded, then, that the inspection of the fracture may be a means of determining the cause of the failure. In this way, possibly, the responsibility may be established between builder and customer as regards pinion mounting.

The complete paper includes a detailed stress analysis which shows the comparative ease with which such a stress problem as the one dealt with can be handled by the photo-elastic method, whereas the use of ordinary engineering methods gives untrustworthy results and the exact mathematical solution based upon the theory of elasticity is impossible.

Acknowledgment is due to the Massachusetts Institute of Technology for permission to use in this article certain of the results included in the thesis submitted by Dr. Paul Heymans, University of Ghent, Belgium, as partial fulfilment of the requirements for the degree of Doctor of Science from the Institute.

Discussion

IN OPENING the discussion, A. L. Kimball, Jr., one of the authors, stated that the importance of the investigation outlined in the paper lay mainly in the determination of perfections of pinion design as regarded stress. The similarity of celluloid to steel in the homogeneity of its elastic properties made it possible to study stress distribution in the celluloid models. Mr. Kimball stated that it was possible to study static stress distribution to gain information about stress due to impact as under both conditions the stress distributed itself in substantially the same way.

In answer to some questions asked by J. O. Madison,[1] Dr. Heymans stated that the celluloid models were mounted on steel shafts and that some of the differences in color indicating stresses in the models were due to variations in age of the celluloid used. Mr. Madison stated that in his practice railway pinions were forced on shafts by first boiling in water, putting them on hot and setting with a given number of blows of sledges of definite weights. E. O. Waters[2] emphasized the possibility of using the photo-elastic method to determine facts relating to stresses, pressure distribution, and wear. Dr. A. A. Adler[3] suggested the rotation of the celluloid models and the securing of pictures under actual operating conditions. He also emphasized the need for solving two-dimensional stresses before considering stresses in three-dimensions.

In this connection Dr. Heymans pointed out that in the three-dimensional elastic problem the stress distribution was not independent of the elastic constants of the material so that any determinations in celluloid might not be applied to other materials with different elastic characteristics.

In a written discussion received subsequent to the meeting Dr. S. Timoshenko[1] pointed out that the photo-elastic method of analysis of two-dimensional problems had a broad application within limits and it was important that these limitations be clearly understood. The method failed where the elastic limit of the material under test was exceeded. Dr. Timoshenko suggested that two observations be made by the authors, including the stress distribution from some external force acting on the tooth of the pinion and the stress distribution for similar pressure acting on the inner diameter. From these observations it was possible to obtain stresses at any point of the pinion by the vector sum provided the resultant stress lay within the elastic limit. He also pointed out that the presence of shafting would cause three dimensional stresses at the common boundary of pinion and shaft, rendering the photo-elastic method invalid. He also pointed out that in a study of the effect of pressure acting on the inner diameter it would have been valuable to have compared the pinion with a ring, the outer diameter of which was equal to the diameter at the root of the teeth.

In a plastic material, such was steel, it was impossible to foretell that a given section would prove weakest at the breaking point of the material, because this section contained the maximum stresses when the body was stressed within the elastic limits. Therefore, the photo-elastic results could not confirm any rupture tests on specimen. Moreover, no conclusions could be reached from the tests themselves. In all, only three were made and it seemed to him that on the basis of these alone, it was just as probable that the breaks through the thicker sections had been caused by the initial stresses in the materials as by the stresses produced by the applied forces.

The present paper would be of direct benefit to the designing engineer if, for any given tooth shape, it gave him a definite relation between the actual stresses across the "Minimum cross-section" of the tooth and those obtained by calculation from the cantilever formula. For example, taking the depth of the tooth above the cross-section considered as equal to $^1/_2$ in., the thickness equal to $^1/_3$ in. and the load at the end equal to 1500 lb., the maximum stress obtained from the cantilever formula would be

$$p_{max} = \frac{1500 \times {}^1/_2 \times 6}{\left(\frac{1}{3}\right)^2} = 40{,}500 \text{ lb. per sq. in.}$$

Comparing this with the stresses 72,600 and 80,000 given by the photo-elastic method, it would be seen that the increase of maximum stress due to the local stresses near the root of the tooth, over that given by the cantilever formula was respectively 79 per cent and 98 per cent.

[1] New York, N. Y.

[2] Asst. Prof. Machine Design, Yale University Jun. Mem. A.S M.E.

[3] Consulting Engineer, New York City Mem. Am.Soc.M.E.

[1] Consulting Engineer, Vibration Specialty Co., Philadelphia, Pa.

WORK OF THE A.S.M.E. BOILER CODE COMMITTEE

THE Boiler Code Committee meets monthly for the purpose of considering communications relative to the Boiler Code. Any one desiring information as to the application of the Code is requested to communicate with the Secretary of the Committee, Mr. C. W. Obert, 29 West 39th St., New York, N. Y.

The procedure of the Committee in handling the cases is as follows: All inquiries must be in written form before they are accepted for consideration. Copies are sent by the Secretary of the Committee to all of the members of the Committee. The interpretation, in the form of a reply, is then prepared by the Committee and passed upon at a regular meeting of the Committee This interpretation is later submitted to the Council of the Society. for approval, after which it is issued to the inquirer and simultaneously published in MECHANICAL ENGINEERING.

Below are given the interpretations of the Committee in Cases Nos. 408 to 410 inclusive, as formulated at the meeting of December 8, 1922, and approved by the Council. In accordance with the Committee's practice, the names of inquirers have been omitted.

CASE No. 408

Inquiry: What thickness is required under Par. M-3 of the Code for Miniature Boilers for the plate forming heads which are not an

(*Continued on page 149*)

MECHANICAL ENGINEERING

A Monthly Journal Containing a Review of Progress and attainments in Mechanical Engineering and Related Fields, The Engineering Index (of current engineering literature), together with a Summary of the Activities Papers and Proceedings of

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The Engineer of Tomorrow

Chas. F. Scott

Dean Kimball in his presidential address last month made a keen analysis of conditions prevailing in our modern life. He showed that the engineer has done his part in the past to bring about the civilization in which we live. The importance of the engineer in present activities was indicated, and he pointed out the larger sphere of action which the engineer should logically take in the future. In public affairs the military leader, the legal leader, and the business or financial leader must give way to the scientific or engineering leader.

Dean Kimball's statements have met with criticism and question. An editorial in a daily paper of one of our large industrial cities says that an engineer is a practical man who sees the things that need to be done and goes ahead and does them, but that he is not fitted temperamentally or by training for political positions of service.

What the engineer will be and what he will do depend upon the engineer himself. The engineering criterion of fitness and quality will be applied and if the engineer does play a leading part in the future, it will be because he is worthy of it.

What type of a man will the engineer of the future be? It is significant that the presidential address was delivered by one of our foremost engineering educators, by one who is directing a factory which is making engineers of the future. Whether his vision of a future in which the engineer will be a national leader will be realized will depend in large measure upon whether Dean Kimball himself and his associates in the engineering educational field are successful in giving men the training and the vision which will fit them for national leadership.

The engineer of tomorrow is the engineering student of today. The place which engineers will take in the future and the part they will play in solving the larger economic, social, and governmental problems which the spread of engineering has created, will depend upon how the engineering student of today develops into the engineer of tomorrow.

The importance of this matter is one which extends far beyond the schools themselves. Engineers, engineering societies, and society at large have a profound concern in the ideals and the training of the engineer. The Society for the Promotion of Engineering Education, of which Dean Kimball is a vice-president, has just organized a Board of Investigation and Coördination for the broad consideration of problems of this kind.

If engineers are to take an important place in the future of this country and of the world we may well ask that a survey be made of the number and type of engineers that will be needed. We should ascertain whether our schools are now equipped for producing the proper number of men of the right kind. If more are needed or if a different type of graduate is called for, then we must have more freshmen, and freshmen of a different type.

The members of this Board are M. E. Cooley, Past-President of the A.S.M.E.; F. W. McNair, President, Michigan College of Mines; D. C. Jackson, Massachusetts Institute of Technology; John H. Dunlap, Secretary, A.S.C.E., and C. F. Scott, Yale University. Dean Bishop, Secretary of the S.P.E.E. is also secretary of the Board. It is interesting to note that three of the members of the Board are members of the A.S.M.E. It is hoped that the Board will later on have funds for a director and staff for the carrying on of the work. One of the policies already adopted is that of securing the active coöperation of the schools themselves in the study of the new problems which are confronting engineering educators. Letters from many engineering schools, now being published in the S.P.E.E. Bulletin, show an active and significant interest in the new movement. The Board also anticipates the coöperation of the various engineering societies and is appreciative of the interest which is being taken by the Mechanical Engineers.

Charles F. Scott.[1]

Engineering Museum Requires Interest and Aid of Entire Profession

The project to establish an engineering museum has attracted wide interest both in the technical and lay press. The matter is now in the hands of a Committee of representatives of the four National Engineering Societies.

Most of the comments on the museum stress its importance as a storehouse for the records of engineering achievement and as an institution affording opportunity for research to the investigator or student. The Secretary of the Smithsonian Institute, in a recent report, concludes that a museum would be a suitable monument showing belated public appreciation of the fact that the commanding place reached by the United States in so short a time is due largely to the full development and utilization of mechanical power in the exploitation of her national resources.

However, a writer in the New York *Tribune* has recently emphasized the high value of such a museum to children in giving them an accurate and inspiring knowledge of mechanical processes. He has a vision of an arrangement of models of the stages of mechanical and industrial developments, a cross-section of the world's mechanical brain, in the various stages of growth, where no boy can spend a day without getting the spark in his soul that may ignite the inventive faculty. This writer's plea is, therefore, for the location of the museum in the midst of the largest number of children, and New York would therefore seem the logical selection. A correspondent has pointed out the industrial greatness of the Middle West and believes that Chicago should be favorably considered. And there are many other diverse ideas on the subject, with the problems multiplying as the interest increases.

The Committee of the Engineering Societies has visited the Smithsonian Institute at Washington and, in coöperation with it, is formulating a plan for a great national museum of engineering and industry similar to the Science Museum at South Kensington, London, the Conservatoire des Arts et Métiers at Paris, and Deutsches Museum at Munich, but suited to the needs of this country. The scheme provides a central institution with branches in different sections and proper exchange facilities.

Ideas regarding the character, scope, and location of the museum and the branches are earnestly solicited by the Committee. Engi-

[1] President, Society for Promotion of Engineering Education.

neers are also urged to preserve the models and records of their original work for eventual deposit in the museum.

Holbrook F. J. Porter is Chairman of the Committee, with offices at the Engineering Societies Building, 29 West 39th Street, New York, N. Y. The representatives of the National Engineering Societies are Edward D. Adams and Charles L. Clarke, representing the American Institute of Electrical Engineers, Frederick A. Delano and Dr. George F. Kunz, representing the American Institute of Mining and Metallurgical Engineers; Clemens Herschel and Nelson P. Lewis, representing the American Society of Civil Engineers; and Reginald P. Bolton and H. F. J. Porter, representing The American Society of Mechanical Engineers.

International Courtesies

THE recent announcement of the appointment of Gano Dunn as Local Honorary Secretary in America of the Institution of Electrical Engineers of Great Britain directs attention to a particularly gracious form of courtesy that has just been shown to the United States by the British. Another example of this is the hospitality extended by the national engineering societies of America to the American Section of the Société des Ingénieurs Civils de France. The privilege of the Library and offices are extended to the American Section. This is reciprocated by similar welcome to members of the engineering societies of the United States who live in Paris.

Such incidents emphasize again the great common interests of all members of the engineering profession which can be broadened by still further coöperative effort especially in research and standardization. The engineer as a professional man is rapidly coming into his own sphere of importance in the maintenance and development of civilization and these international contacts will be of great assistance in maintaining a uniform front in engineering activity.

International Engineering Congress, 1926

ON TUESDAY, January 9, in the Engineering Societies Building in New York, a conference of engineers most experienced in matters relating to engineering congresses was held to discuss the plan, scope, and method of financing of a proposed International Engineering Congress to be held in Philadelphia in 1926 in connection with the Sesquicentennial of the Declaration of Independence of the United States. The movement for this proposed Congress was initiated by the Engineers' Club of Philadelphia, which invited the leading engineering societies to join with it on a committee to formulate a plan.

The secretaries of the engineering societies proposed a plan of organization for an International Engineering Congress which was subsequently approved by those societies and adopted at a meeting of the temporary organization committee held in Philadelphia last summer. Under this plan, a board of management was appointed to organize and conduct a congress under the sponsorship of the engineering societies, and the board appointed, among other committees, a committee on plan and scope. This committee was the body which conducted the public hearing being announced here.

The hearing was virtually an exchange of experiences between those competent to speak on the subject, and among those who addressed the meeting were Messrs. A. M. Hunt, H. Foster Bain, Henry A. Lardner, Fred Lavis, A. R. Ledoux, John W. Lieb, Elmer A. Sperry, M. E. Cooley, Charles F. Rand and F. M. Feiker.

The subjects discussed were scope of the Congress, finances, relation to meetings of engineering societies, relations with governmental agencies, and publicity.

Patent Office Salaries and the Public Interest

THE individual members of the Engineering Societies by giving their powerful aid in securing the enactment on February 18, 1922, of the Lampert Patent Office Bill, H.R. 7077, helped to stop the disintegration of the Patent Office by constant resignations and prevented its complete collapse. Without this help it would soon have been swamped and ceased to function usefully.

The Lampert Bill raised the salaries of the Patent Office examiners to $3900 per year and stopped the resignations in the upper grades, where it was most important to stop them. However, the increase from $2750 to $3900 was by far the largest proportionate increase that had ever been obtained through any bill in Congress, and was all that it was possible to obtain at one time. The increased salary is not sufficient, however, to attract men who are highly qualified for the work and to induce them to make a life career of the position, as they must do in order to reach their highest efficiency—in such numbers as to supply all or nearly all of the examining divisions.

The work of an examiner is of great importance to the public interest. By having examiners with sufficient scientific and judicial qualifications and sound judgment and adequate personality, patents will be granted wherever the inventions warrant it, and not be refused because the distinctions of the prior art, although important, are not easily discerned. Such examiners will also with sound judgment detect those cases where the distinctions between the alleged invention and the prior art are not really practical commercial distinctions, but are mere paper differences, and will thereby prevent the granting of patents which can but result in useless litigation that is expensive not only to the patentee and his backers but to the innocent defendant and the Government in the waste of time of the courts.

The inventions which examiners must pass upon are often of great immediate value, and their ultimate value, through their permanent addition to the public domain, are beyond calculation. Therefore, the administration of the Patent Office under examiners of the high type mentioned will increase the market value of patents and thereby stimulate the production of inventions in general—to the great and permanent benefit of the American public.

It is believed that a salary of $5000 for a primary examiner would attract sufficient men of the type described to fill that position in practically all of the examining divisions. There is an excellent opportunity to obtain that salary for the position by aiding in bringing about the enactment of the Sterling-Lehlbach Bill for the Reclassification of Governmental Positions and Salaries, H.R. 8928. This bill has passed the House by a large majority, but in doing so the salary for the position of primary examiner was reduced from $5040 to $4600. The Civil Service Committee of the Senate has reported the bill recommending a restoration to $5040. As the bill affects appropriations, it was also referred to the Appropriations Committee of the Senate, and has been kept there nearly a year by the desire of Senator Reed Smoot to substitute another bill which, while having a different scheme of classification, has substantially the same schedule of salaries.

As the Sterling-Lehlbach Bill has already passed the House, it would obviously be much easier to enact than a substitute bill. The information is that if sufficient public interest is shown, a compromise not affecting the salaries could probably be brought about and the Sterling-Lehlbach Bill reported to the Senate; and if that is done the Bill can be passed through the Senate without great difficulty, after which it seems likely that the House can be induced to agree to the larger salaries.

The Bill has been approved by the following organizations:

The New York Patent Law Association
The National Federation of Federal Employees
The American Federation of Labor
The Federated American Engineering Societies
The American Association of Engineers
The National Civil Service Reform League, and others.

The engineers, chemists, scientists and manufacturers proved with the Lampert Bill that they could induce Congress to pass a just and wise measure in the face of intense opposition. There is no such opposition to the present bill as there was to that bill. Each member of each of the Engineering Societies is asked to write to Senator Smoot and to one of his own senators urging the immediate enactment of the Sterling-Lehlbach Reclassification of Salaries Bill, H.R.8928. If every member will do his duty, it seems more than probable that the bill can be passed and the Patent Office placed on that high plane which its great usefulness warrants.

The enactment of this bill would raise the standard of all professional service under the Government as well as that of Patent Office examiners. Let us all pull together and finish the work which we have so successfully begun.

Edwin J. Prindle.[1]

[1] Chairman, Patents Committee, American Engineering Council.

Dr. Millikan Awarded Edison Medal

THE Edison Medal for 1922 has been awarded to Dr. Robert A. Millikan, of Pasadena, Cal., for "his experimental work in electrical science." The experiments for which he is best known are those dealing with the isolation of the electron.

After receiving the degrees of A.B. and A.M. from Oberlin College Dr. Millikan began teaching physics at the University of Chicago, becoming professor of physics in 1910 and remaining in that capacity until 1921. Since that time he has been director of the Norman Bridge Laboratory of Physics and chairman of the administrative council of the California Institute of Technology. He was granted a Ph.D. from Columbia in 1895 and holds degrees from several other universities in this country and Germany.

Dr. Millikan has written a number of books on physics, besides contributing to the technical press on this subject. He has rendered valuable service as vice-chairman of the National Research Council since 1917 and was a member of the board of editors of the *Physical Review* from 1911 to 1914. In 1913 he was awarded the Comstock prize for research in electricity. Recently Dr. Millikan has been conducting experiments to bridge the gap between light and X-ray phenomena.

Among other recipients of the Edison Medal have been Elihu Thomson, George Westinghouse, Alexander Graham Bell, Benjamin G. Lamme, W. L. R. Emmet, Michael I. Pupin, and C. C. Chesney.

I.E.E. Appoints Gano Dunn Honorary Secretary for United States

THE COUNCIL of the Institution of Electrical Engineers, of Great Britain, has appointed Gano Dunn, president of the J. G. White Engineering Corporation, New York City, honorary secretary of the Institution for the United States, to succeed the late G. G. Ward.

Mr. Dunn is an eminent figure in engineering and will be able to render the Institution valuable service as its local secretary. His association with the J. G. White Engineering Corporation dates from 1913, and for two years previous to that he was vice-president in charge of engineering of the J. G. White & Co., Inc. He was born and educated in New York City, receiving the degree of B.S. and M.S. from the College of the City of New York, and E.E. and honorary M.S. from Columbia.

As a member of numerous technical societies and organizations Mr. Dunn has a wide acquaintance among the engineers of this country. He has served as an officer at two international electrical congresses, the one at St. Louis in 1904 and that at Turin in 1911. He was a delegate to the Second Pan-American Scientific Congress in Washington, 1915, and has served on the Nitrate Commission of the War Department and on the Engineering Committee of the Council for National Defense. He has been a member of the International Electrotechnical Commission, vice-chairman of the National Research Council, chairman of Engineering Foundation, and president of the A.I.E.E., the U.E.S., and the John Fritz Medal Board of Award. He belongs also to the National Academy of Sciences, the Association of Iron and Steel Electrical Engineers, the American Society of Civil Engineers, The American Society of Mechanical Engineers, and The Franklin Institute, as well as to many other technical and fraternal organizations.

New Honors for Colonel Dwight and Mr. Rand

"AN EVIDENT manifestation of the good feeling which exists between the technical men of South Africa and those of the United States" is the light in which the American Institute of Mining and Metallurgical Engineers, in its journal, *Mining and Metallurgy*, views the election of its president, Col. Arthur S. Dwight, to honorary membership in the Chemical, Metallurgical and Mining Society of South Africa. In the conferring of honorary membership upon Charles F. Rand, chairman of the Engineering Foundation, by the Association of Members of American National Engineering Societies in Cuba, a similar internationality of engineers is seen. Both of these engineers have been previously honored by foreign countries, France bestowing upon each the Cross of the Legion of Honor for distinguished service to France and to civilization, and King Alfonso decorating Mr. Rand for his achievements in the development of Cuban iron mines.

Dr. Rosenhain to Lecture in Eastern States

DURING February and March Dr. Walter Rosenhain, head of the metallurgical department of the National Physical Laboratory at Teddington, England, will tour the eastern states, lecturing before various universities and technical organizations on metallurgical subjects. Dr. Rosenhain comes to this country under the auspices of the Institute of Metals Division of the American Institute of Mining and Metallurgical Engineers, and will deliver the second annual Institute of Metals lecture, on Solid Solutions, before that body on February 19. Subjects of other lectures will be Hardness and Hardening, The Structure and Constitution of Alloys—to be delivered before Franklin Institute on February 15—Strain and Fracture in Metals, Aluminum Alloys, and Metallurgical Research at the National Physical Laboratory. His itinerary will include Lehigh, Columbia, and Yale Universities, Production Club (Waterbury, Conn.), Massachusetts Institute of Technology, Case School of Applied Science, American Society for Steel Treating (Detroit), University of Illinois, and Engineers' Club of Dayton.

The Charles A. Coffin Foundation

THE General Electric Company, as an expression of appreciation of the constructive service which its founder, Charles A. Coffin, has rendered the electrical industry, has established a fund of $400,000, the income from which is to be used to encourage and reward similar services.

Mr. Coffin, who has been identified with the development of the electrical industry since 1882, founded the General Electric Company and was its leader for thirty years. He retired from active business in May, 1922. The Charles A. Coffin Foundation, created at that time by the Board of Directors of the General Electric Company, is under the direction of a Foundation Committee appointed by the board. The income from the fund recently set aside by the company will be distributed annually in four ways, as follows:

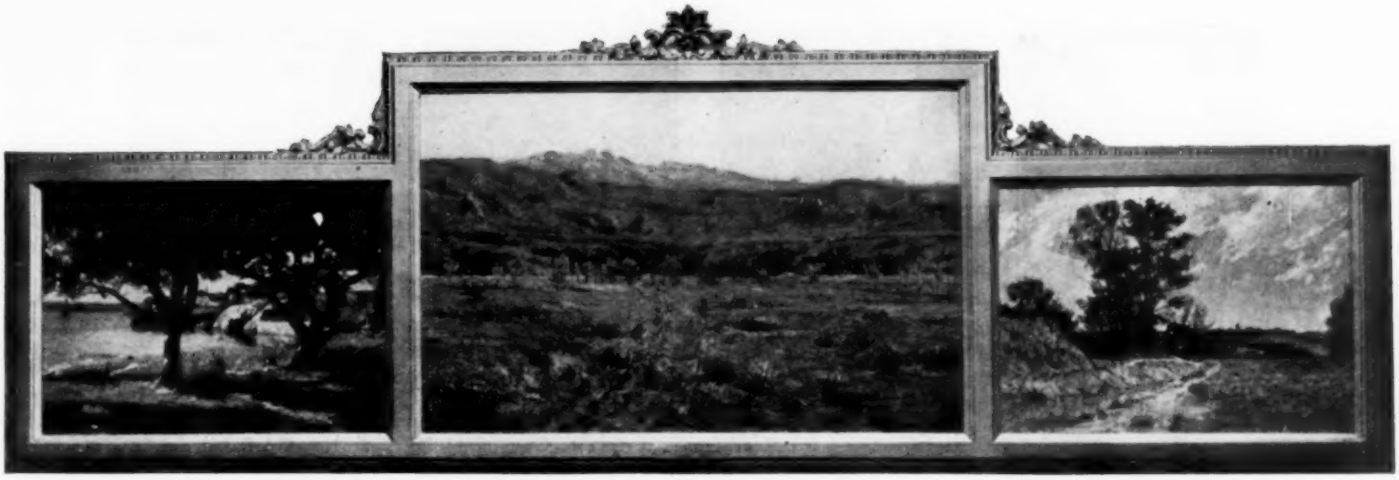
1 Eleven thousand dollars in prizes for the most signal contributions by employees of the General Electric Company toward the increase of its efficiency or progress in the electrical art

2 A gold medal to the public-utility operating company within the United States which, during the year, makes the greatest contribution toward increasing the advantages of the use of electric light and power for the convenience and well-being of the public and the benefit of the industry. The company receiving the medal will also receive one thousand dollars for its employees' benefit or similar fund

3 A gold medal to the electric railway company within the United States which during the year, makes the greatest contribution toward increasing the advantages of electric transportation for the convenience and well-being of the public and the benefit of the industry. The company receiving the medal will also receive one thousand dollars for its employees' benefit or similar fund

4 Five thousand dollars to graduates of American colleges and technical schools who, by the character of their work, and on the recommendation of the faculty of the institution where they have studied, could with advantage continue their research work either here or abroad; or some portion or all of the fund may be used to further research work at any of the colleges or technical schools in the United States. The fields in which these fellowships and funds for research work are to be awarded are electricity, physics, and physical chemistry.

The progress report of the United States Coal Commission which was issued January 15, is the first tangible result of this experiment in basing important economic legislation upon careful investigation. John Hays Hammond, chairman, and George Otis Smith are the engineers on the Commission. E. E. Hunt, a member of the F.A.E.S. Committee on Elimination of Waste is Secretary to the Commission.



Art and the Engineer

VISITORS to the Engineering College of the University of Cincinnati are struck with the artistic atmosphere of the buildings. In place of the starkly framed photographs of machines and locomotives which usually add to the scholastic and technical atmosphere of an engineering school, they find there paintings in oils and water colors of a remarkably high order. Glances at the title plates on the pictures disclose the fact that the paintings in most cases have been presented to the University by the students themselves.

This interest in art at the Engineering College of the University of Cincinnati is of long standing. It has been fostered by Dean Schneider, who believes that the engineering student should be stimulated in art, music, and literature, not only because of the enjoyment and relaxation which an appreciation of art affords, but also because a student well grounded in cultural subjects makes a better engineer. It is possible to introduce him to the liberalizing influences of history and literature through the regular college courses, but a sustained interest in art can be secured only by furnishing him with actual and more or less constant contact with the artistic.

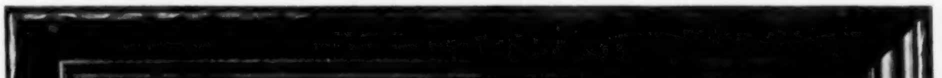
The transformation of the corridors into an art gallery was started in 1916 with the mural decoration of the Library. Upon graduation each class gives a fund varying from six hundred to twelve hundred dollars, while various groups, such as student branches of the national engineering societies, have donated smaller amounts. These gifts have been entirely voluntary and have resulted in a collection of more than fifty paintings, only six of which have not been given by the students.

To maintain the excellence of the collection, the standard for permanent hanging in the Cincinnati Art Museum has been adopted for pictures in the College of Engineering. The Director and Curator of the Museum are consulted in the purchase of pictures. This high standard, and the fact that the pictures become the property of the city of Cincinnati to be hung in perpetuity in a fireproof building of the University, have secured the coöperation of the artists, with resultant benefit to the collection.

The collection is worth study by any engineer who is within reach of it, whether he be an art connoisseur or not. The interest it arouses among students, apparent continually as one passes through the corridors, is ample proof that it must be effective in the lives and works of the engineers who graduate from the University. The competitive system of industry in America has left little time for technical men to study and appreciate the real economic value of culture. So rapid has been our industrial growth that the ideal and artistic aspects of industry have sometimes been forgotten.

Engineers, however, have recognized for some time that wall paper and bridges should be beautiful as well as utilitarian—that machinery should always be the servant of man and, wherever possible, the servant of art. The Engineering College of the University of Cincinnati is graduating men who have learned beauty of line and color through actual contact with it, and who carry with them into the business and industrial world an appreciation of art which is entirely in keeping with the dignity of the profession they have entered.

To fill properly his place, the engineer of tomorrow should be well-rounded in painting and architecture, the arts intimately associated with the guiding ideals of permanence and beauty.



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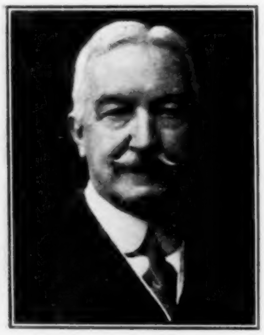
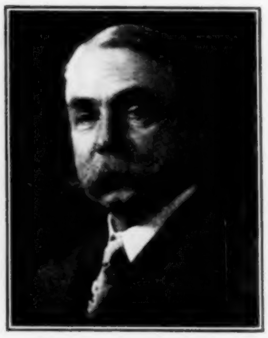


Mortimer E. Cooley
President



P. N. Moore
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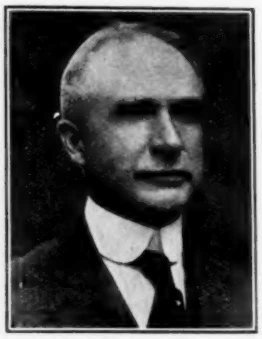
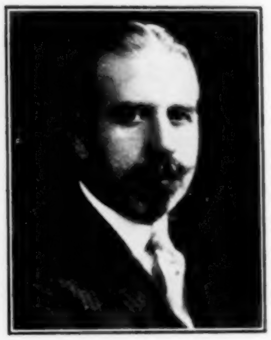


Calvert Townly
Vice-President



H. E. Howe
Treasurer

Newly Elected Officers of American Engineering Council, Governing Body of the F.A.E.S.

Caetani Welcomed by Engineering Council

Italian Ambassador States Need for Engineering Logic in Public Affairs. Enthusiasm and Action Mark Second Annual Meeting of F.A.E.S. Dean Cooley Re-elected President. Constitution to Be Revised. Notable Resolution Adopted Stating Need for Broader Engineering Training

INTENSITY of purpose and enthusiasm marked the Second Annual Meeting of the American Engineering Council, the governing body of the Federated American Engineering Societies, which was held in the Cosmos Club, Washington, D. C., for two days, January 11 and 12, 1923. The climax occurred during the dinner on the opening evening of the session, at which the guest of honor and principle speaker was the Ambassador from Italy to the United States, Prince Gelasio Caetani, an American engineering graduate and for thirteen years engaged in engineering work in this country. Following the meeting of the Council, the executive Board met to organize the work for the current year.

The Dinner

Dean M. E. Cooley, re-elected President of the Council during the proceedings of the day, presided over the dinner at the Chevy Chase Club. The first speaker of the evening was the Honorable John James Tigert, United States Commissioner of Education. Mr. Tigert is an able orator and his address was a graphic picture of the popular conception of the engineer. He was followed by Secretary Calvin W. Rice, of the American Society of Mechanical Engineers who gave a résumé of his visit to South America where he represented a number of engineering and civic bodies at the International Engineering Congress and the Centennial Celebration of Brazil at Rio de Janerio. Afterwards he visited a number of South American countries and returned with the firm conviction that there is a remarkable opportunity for engineering coöperation with South America. Mr. Rice's inspiring report appeared in the January issue of Mechanical Engineering.

The third speaker was Dr. Elmer A. Sperry who has just returned from a trip to Japan where he presented letters of congratulation from American engineering societies to the semi-jubilee celebrations of two Japanese engineering bodies. He in turn emphasized the necessity for developing the most intimate relations with Japanese engineering organizations.

The messages of international coöperation between engineers and the sentiment expressed by Mr. Rice in his talk that an engineer visiting a foreign country is the most potent ambassador of peace and goodwill was clearly in the minds of those present when Prince Gelasio Caetani rose to begin his address. His fellow engineers gave him a sincere and warm welcome to the country where he had spent the best year of his life, leaving it only at the call of duty to perform feats of wisdom and valor for his own country in the last war. He returns as the representative of the Mussolini government and those who were privileged to meet him and hear him were convinced that his stay will be fruitful to both Italy and the United States. His personal charm and his sincerity held his audience and at the close of his speech he was given an enthusiastic ovation.

Address of Prince Gelasio Caetani

Your kind invitation to be a guest at the annual banquet of the Federation of American Engineering Societies reached me while I was preparing to leave for the United States. I read it with deep satisfaction for it made me feel that, besides sailing for America as Italy's Ambassador, I was going home to my old stamping grounds somewhat still invested with the qualifications of an engineer.

We pride ourselves in saying: "Once an engineer, always an engineer." Whatever may be the course of life followed by any of us, it will always be marked by the indelible seal of the scientific, practical and logical training to which an engineer is subjected during the early years of life.

Some have made the remark in criticism that engineers lack political intuition and ability; I would answer that a larger dose of logic and positiveness applied to politics would bring great advantages to public affairs.

Between Italy and the United States there has never existed political rivalry or serious commercial competition; our relations have been confined almost exclusively to contacts of labor, of engineering, of commerce, of science and of art. Much can be accomplished to the mutual advantage of our peoples; a large share of the success will depend upon the coöperation of the engineers and I know that I will always be able to rely on your goodwill.

I do not hesitate to state that Italy and the United States are at present the most youthful nations of the world. Italy is the oldest one in history and three times has ruled the world; once politically, once spiritually and once intellectually. However, as a political and social unit Italy did not exist from the fall of the Roman Empire to the middle of last century; as race and as nation it had an enforced rest of some fourteen centuries. With the forming of its national unity in 1870 it awakened to a new life; born again as a new being to play its role in world's history, it is healthy, fertile and exuberant of youthful energies. The best proof of this is given by the latest events which led to the establishment of a new national government. The younger and healthiest part of the people, the bulk of the nation, openly rebelled against the old ways which were leading Italy into a critical condition; not only bolshevism and anarchy have been wiped off the map, but also demagogy and all low grade politics aiming to the fostering of party and class interests. In an amazingly short time a complete change of all orders of things was effected, not a miraculous revolution, as some people have thought to style it, because the events were nothing more than a sudden realization of what had been the deep desire and the will of the people to which circumstances so far had not allowed expression. An event that within 24 hours raises the quotations of government securities and restores peace and general confidence, without a drop

of blood being shed or a windowpane broken, cannot be styled a revolution.

The other youngest nation in the world, I was saying, is the United States, the new great power of the history to come; unlimited in its financial powers, unrivaled in its capacity of organization and technical knowledge, wonderful in the possibilities of its vast empire. The coöperation of these two young countries will lead to remarkable results; both our people are laborious and have an inventive, engineering turn of mind.

Italy's largest asset is the remarkable quality of its people's labor; sober, intelligent, hardworking and plastic the Italian peasant or workman will in an incredible short time become efficient in whatever he is called to do.

Of all this many Americans are perhaps not yet fully aware for the reason that the wave of Italian emigration which shortly preceded the war was so sudden that it was not utilized to best advantage.

Much however is still to be done in Italy itself; its resources are far from being fully developed and there are many opportunities for American capital, machinery and technical organization to be usefully applied in Italy.

The electric industry in our country has made rapid strides and as to percentage of utilized water power, Italy ranks, I believe, foremost in the world. Its use results in an economy of about two billion lire, otherwise necessarily spent on fuel imports.

In 1898 the electric energy developed in Italy amounted only to 87,000 kw.; it increased to 426,000 kw. in 1908, to 1,240,000 kw. in 1918 and power plants for some other 1,000,000 kw. are planned or under construction. About 800,000 kw. are still to be developed. On the Tirso in Sardegna a reservoir of 416 million cubic meters capacity is being constructed. It will be the second largest in the world, ranging immediately after the Assuan dam and will develop some 50 million kw-hrs., and irrigate 60,000 acres of land. In the Trentino a reservoir is to be constructed on the Noce torrent of 180 million cubic meters capacity. The dam, 400 feet high, will be practically a concrete wedge driven in a mountain gully, measuring less than sixty feet wide at the base and only 100 feet wide at 300 feet above the bottom. In Southern Italy the large reservoir in construction on the Sila mountain will develop over 110,000 kw. and irrigate large tracks of fertile land. Another interesting plan which is gradually being carried through is to connect the northern power plants, fed by the summer streams of the Alps, with those of central Italy where water is plentiful in winter and rather poor in summer, by a network of high tension lines and by standardization of voltage to obtain a better seasonal compensation than could be secured by the use of even very large reservoirs.

Railroads are to be electrified and telegraphs and telephones are to be reorganized then gradually handed over to private enterprises; experience has proved that state administration of industrial concerns ends always in a financial and technical failure.

I should mention also the large works for reclaiming waste or marshy land by irrigation or drainage. There are 148 enterprises of this kind in Italy for the reclamation of some three million acres of land; of these 35 have been completed covering an area of about 820,000 acres.

My experience in the United States has been my most valuable asset in life. Before leaving Rome, at a dinner given to me by the Italian engineers, I exhorted the young engineers to get a few years of practical training in America. I hope you will do likewise by encouraging your students and graduates to spend some time in Italy as nothing broadens the mind more than to breathe an atmosphere different from that of one's own town and country. Italy's atmosphere is vibrating with wonderful reflexes of a long and glorious past and full of promise for a remarkable future. It is a great art in life to single out and appreciate other people's good qualities and try to make them your own. Similar intercourse between our young students who, in a few years, will be the active men of our countries, will be a powerful factor in reaching the principal and ultimate aim I will have in view in carrying out my duty as Ambassador, that is to strengthen the bonds of friendship and esteem between Italy and the United States.

The Business Session

The opening event at the first business session on January 11 was the presentation of a report by President Cooley, who told of his visits to many local engineering societies during the year. He called for the support of each member of the Council in interesting and inspiring the engineers of the country in the tremendous problems that are facing present day civilization.

Secretary Wallace then related the progress that had been achieved in the work of the Council during the past year. In addition to the routine matters handled by the executive office, Mr. Wallace told of the international interest expressed in the report on Waste in Industry and the Twelve-Hour Shift in Industry. He stated that the Council was firmly established in the mind of official Washington as the representative of a profession that was striving to give disinterested service for the common good. He related the efforts in presenting the facts about a national hydraulic laboratory for the study of river flow and the successful outcome of the struggle on the floor of the House of Representatives for an increased appropriation for topographic mapping. He pointed out however, that while the general information among engineers about the Council and its work was very much better during 1922 than in preceding years, still there was a very severe problem of conveying to every engineer in the United States the importance, the significance and the potentiality of the Federated American Engineering Societies and the American Engineering Council, its governing body.

The report of the treasurer, W. W. Varney, showed a balance of about five hundred dollars for the fiscal year of 1922. Later in the day a budget for the year 1923 of $45,000 was approved. The representatives of the local societies signified their willingness to pay the full assessment of one dollar per member instead of accepting a reduction voted.

Election of Officers

Under the interpretation of the Constitution that the ineligibility of a president for re-election does not apply when the present incumbent is filling out an unexpired term, Dean Mortimer E. Cooley was re-elected as president of the American Engineering Council for a term of two years. Dean Cooley is a member of the delegation of The American Society of Mechanical Engineers. Calvert Townley of the American Institute of Electrical Engineers Philip N. Moore, of the American Institute of Mining and Metallurgical Engineers and Gardner S. Williams of the Detroit Engineering Society were elected as vice-presidents. Mr. Williams was chosen to fill the unexpired term of Dean Dexter S. Kimball of one year and the others will serve the term of two years. Dr. Harrison E. Howe, of the American Institute of Chemical Engineers was elected Treasurer.

Publications

Several times during the meeting the question of the scope and character of a publication for the Council was taken up and discussed. The representatives of the local engineering societies were convinced that a publication was vitally essential and seemed in agreement that it be frequent and of small size. Many expressed the opinion that a publication was the answer to the problem of getting and holding the interest of the individual member of the constituent bodies. The matter was thoroughly discussed and was left to the executive board for the determination of a policy. The present monthly bulletin is to be continued and a resolution was passed which will make it possible to secure second-class mailing rates. The Council also favored the suggestion that a joint news publication of the national engineering societies and the F.A.E.S. be established.

Revision of Constitution

The Council acted favorably on the recommendation of the Executive Board that a committee be appointed to revise the constitution of the Federated Engineering Societies. In the discussion of this motion it was apparent that the members of the Council were in agreement that this be no mere editing but a thorough revision based on a study of the fundamental principles that have been established during the two years of the Council's operation. One action of the Council which may have some effect on the revision of the governing laws was the passage of a resolution that advice and assistance be solicited from each constituent body in so far as this can be done without delaying action.

Two revisions of the by-laws amounting only to interpretations were passed and one change relating to allied technical organizations in foreign countries was laid on the table. It was also decided to provide by constitutional amendment for alternates both for members of Council and for the Executive Board. The consensus of sentiment favored the practice of alternates to the Executive Board being members of the Council.

Government Activities

Several very important actions of the Council related to activities of the Government.

The Council passed the resolution proposed by a committee of the Council at the request of the Boston Society of Civil Engineers recommending that the administration of the Federal Water Power Act be rendered more effective by the maintenance of a permanent, adequate, trained personnel solely for this purpose.

A second resolution recommended that the Bureau of Foreign and Domestic Commerce be authorized to collect and disseminate facts relating to the operation of business.

Compensation and classification of government employees, especially engineers and patent examiners, was given long and careful consideration. The Patents Committee pointed out the need for high grade examiners in the Patent Office which the present government regulations will not permit. The importance of uniform rates of pay for similar kinds of work in different departments was also emphasized. Accordingly, the Council passed a resolution recommending to Congress that adequate rates be provided for the technical employees of the government.

Reforestation

The Committee on Reforestation reported a program by which the entire engineering profession and the public will be educated as to the need for immediate action in reforestation. The perpetuation of the timber supply of the country is a problem that vitally affects, not only the industries that use lumber but also the water supply and the water-power of the future.

Industrial Ideals and Education

Perhaps the most important pronouncement of the Council during the recent meeting was the resolution proposed by the Committee on Industrial Ideals. This resolution relates primarily to the increased present day responsibility of the engineering educator. Many times in discussion of other topics, this responsibility was emphasized and after the passage of this resolution, Professor C. F. Scott, a delegate of the American Institute of Electrical Engineers and President of the Society for the Promotion of Engineering Education told of the remarkable plan developed by the last named society for the preparation of a statement of the fundamentals of engineering education, vital if the engineer is to undertake his great responsibility in the maintenance and development of civilization.

The resolution follows:

Resolved That: The American Engineering Council by action at its meeting on January 11, 1923, desires, on behalf of the Federated American Engineering Societies, to bring to the attention of the engineering colleges throughout the country the need of pointing engineers toward leadership in public affairs.

For a century engineers have directed their energy toward the utilization of the physical forces and the materials of nature. The developments which they have brought about have created an epoch in human history. While these developments have been of inestimable benefit, and modern society could not exist without them, they have introduced many public problems and social readjustments, so closely related to the engineer's activities, that it is increasingly evident he must assume an active part in their solution.

Recognizing this growing need, the engineers of the country formed the Federated American Engineering Societies, primarily to place their knowledge and training at the public service on all public matters affecting engineering, or affected by it.

Engineering education, reflecting closely the attitude of engineers heretofore, has confined its work almost exclusively to scientific and technical training, giving little, if any attention to the social and human aspects of engineering enterprises The American Engineering Council therefore, speaking for the engineering profession, urges upon engineering colleges an increased attention to the social aspects of engineering activities, and a broadening of their technical training, in every way possible, to develop in engineering students the spirit of, and a capacity for, active leadership not only in industry, but in public affairs.

Wellman Memorial Meeting

A notable meeting in honor of Samuel T. Wellman, Past-President A.S.M.E., was held in Cleveland, Ohio on Tuesday Evening January 9th, by the members of the local sections of national engineering societies coöperating with the Cleveland Engineering Society in a series of meetings on the "Iron and Steel Indystry," which are being given by the Associated Technical Societies of Cleveland. The Sections coöperating were:

The American Association for Steel Treating,
The American Chemical Society,
The American Society of Mechanical Engineers,
The American Institute of Mining and Metallurgical Engineers,
The American Association of Iron and Steel Electrical Engineers.

The attendance was large and among those present were many of the prominent engineers of Cleveland and the vicinity, especially the older men who had known Mr. Wellman and his achievements from personal observation over a period of many years that they had together struggled in not only the development of the steel industry but also of the many devices originated in Cleveland, which have contributed so much to the success of the iron and steel industry in America. Mr. Wellman was Past President and also an Honorary Member of the Cleveland Engineering Society.

The meeting was presided over by Mr. F. L. Sessions, Chairman of the Cleveland Section of the American Society of Mechanical Engineers.

The first speaker was Mr. James H. Stratton, a former associate of Mr. Wellman and Engineer of Construction at the Wellman-Seaver-Morgan Co., Cleveland, Ohio. Mr. Stratton in his very fitting remarks regarding Mr. Wellman, stated that one of the best records we have of his achievements were written by Mr. Wellman himself for use in his presidential address[1] before the Annual Meeting of the American Society of Mechanical Engineers in December, 1901.

The first speaker was Ambrose Swasey who had been a life-long friend of Mr. Wellman. Mr. Swasey related many incidents that occurred in their extended travels together which displayed the remarkable qualities of the man. In closing, Mr. Swasey said:

"Mr. Wellman built along broad lines, with noble purposes. He gathered himself friends who were the captains of the steel industry, such as Alexander Holley, Captain Jones of Pittsburgh, and John Fritz. It was Mr. Wellman who called a number of prominent engineers and manufacturers together to give consideration to the celebration of Mr. Fritz's eightieth birthday. The outcome of that conference was a dinner held in the fall of 1902 to celebrate that birthday and at that dinner, the John Fritz medal was established, the greatest honor in the hands of the engineers of America."

The splendid, exceptional steel which Mr. Wellman produced, the wonderful mechanisms which he designed and constructed and sent out into the world, contained those same genuine, dependable qualities which Mr. Wellman possessed to such a remarkable degree. And how true are the words of the young poet Longfellow as regards Mr. Wellman's life and work, "We potters make our pots of what we potters are."

Mr. Clifford C. Smith paid a tribute to Mr. Wellman's foresight shown in his work of improving and developing the basic open-hearth process, the open-hearth charging machine, the ingot charger for the blooming mill and the lifting magnet for handling steel. He also testified to Mr. Wellman's knowledge of paintings, pottery and photography and architecture and the manner he stimulated admiration for these finer things in all those with whom he had contact.

The other speakers were John McGeorge, E. Francis, F. Moeller, A. D. Hatfield and W. G. Hildebran all of whom had been closely associated with Mr. Wellman in various capacities. They related many of the intimate incidents of his life that showed his all around greatness.

In later meetings of the year, the Cleveland Sections of the National societies will discuss the technical phases of the iron and steel industry. Each section will have charge of one meeting.

[1] Transactions, A.S.M.E., vol. 23, p. 78.

Standards for Herringbone Gears for General Commercial Use

Proposed by the A.E.S.C. Sectional Committee on the Standardization of Gears

THE May, 1922, issue of MECHANICAL ENGINEERING contained reprints of the first five gear standards prepared by the Sectional Committee on the Standardization of Gears for which the American Gear Manufacturers Association and The American Society of Mechanical Engineers are joint Sponsors. These five standards having been carefully revised as a result of the criticisms and suggestions received are now before the Sponsor Bodies for final approval and submission to the American Engineering Standards Committee as Tentative American Standards.

At the September meeting the Committee put the finishing touches to the sixth of the gear standards to be completed, and it has recently presented to the Sponsor Bodies this proposed standard for Herringbone Gears for General Commercial Use. This standard is printed below in full for the information of the readers of MECHANICAL ENGINEERING and also to enable them to criticise and comment on this proposal to the Sponsor organizations. All communications should be addressed to Mr. C. B. LePage, Assistant Secretary, The American Society of Mechanical Engineers, 29 West 39th Street, New York.

STANDARD TOOTH FORM

1 *Pressure Angle.* The pressure angle in plane of rotation of gear equals 20 deg.

NOTE: The angle of the cutting tool in the normal plane of the gear at the pitch is 18 deg. 31 min.

2 $\textit{Addendum} = \dfrac{0.8}{P}$

NOTE: P = diametral pitch in plane of rotation.

3 $\textit{Dedendum} = \dfrac{1}{P}$

4 $\textit{Clearance} = \dfrac{0.2}{P}$

5 $\textit{Whole Depth} = \dfrac{1.8}{P}$

6 *Helical Angle.* The helical angle of teeth with axis equals twenty-three (23) degrees.

NOTE: The helix angle of a helical gear is the angle between the helix at the pitch diameter and a plane which contains the axis of the gear.

7 *Enlargement for Pinions.* The enlargement for pinions with 17 teeth or less is given in Table 1.

NOTE (*a*): Enlargement for any given pitch is found by dividing the value given for 1 pitch for the number of teeth desired, by the pitch required.

NOTE (*b*): Calculate the outside diameter as usual, then add the amount of enlargement to the pinion and subtract the same amount from the gear. Use only when pinion has 17 teeth or less.

GEAR-BLANK DIMENSIONS

8 $\textit{Pitch diameter} = \dfrac{N}{P}$

NOTE: N = number of teeth.

9 $\textit{Outside diameter} = \dfrac{N + 1.6}{P}$

TABLE 1 ENLARGEMENT FOR PINIONS

No. of Teeth	Min. No. of Teeth in Melting Gear	Diametral Pitch 10	8	6	5	4	3 1/2	3	2 1/2	2	1 3/4	1 1/2	1 1/4	1
7	28	.1181	.1478	.1976	.2364	.2954	.3382	.3938	.4726	.5906	.6762	.7882	.945	1.181
8	27	.1064	.133	.1782	.2128	.2662	.3046	.3548	.4258	.5322	.6094	.7102	.8514	1.064
9	26	.0947	.1184	.1586	.1894	.2368	.2712	.3158	.379	.4736	.5424	.6322	.7578	.9471
10	25	.083	.1038	.1392	.1662	.2076	.2378	.2768	.3322	.4152	.4756	.5542	.6642	.8302
11	24	.0713	.0892	.1196	.1428	.1784	.2044	.2378	.2854	.3568	.4088	.4762	.5706	.7132
12	23	.0596	.0746	.1002	.1194	.1492	.171	.1988	.2386	.2982	.342	.3982	.4772	.5962
13	22	.0479	.06	.0806	.0958	.12	.1376	.1598	.1918	.2398	.2752	.3202	.3836	.4792
14	21	.0362	.0454	.0612	.0726	.0906	.1042	.1258	.145	.1812	.2082	.2424	.29	.3623
15	20	.0245	.0308	.0416	.0492	.0614	.0708	.0818	.0982	.1228	.1414	.1644	.1964	.2453
16	19	.0128	.0162	.0222	.0258	.0322	.0374	.0428	.0514	.0642	.0746	.0864	.1028	.1283
17	18	.0011	.0016	.0026	.0024	.0028	.004	.0038	.0046	.0058	.0078	.0084	.0092	.0113

10 *Active Face.* $\text{Minimum width of active face} = \dfrac{1.6}{P}$

NOTE: Total face is the width of the gear including the groove. Active face equals the total face minus the width of the groove.

11 *Groove.* The width and depth of groove in center of blank for gears cut with hobs at right angles to the axis of the gear are given in Table 2. The letters A, B, and C in the table refer to Fig. 1.

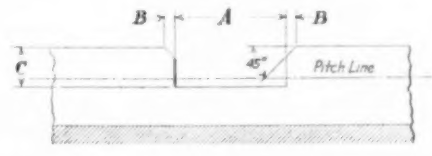


FIG. 1

TABLE 2 DIMENSIONS OF GROOVE

Diametral pitch, inches	A, inches	B, inches	C, inches	No. of threads in hob
10	7/16	1/64	1/8	5
8	7/16	1/64	1/8	4
7	1/2	1/64	1/8	4
6	3/4	1/32	5/32	4
5	3/4	1/32	3/16	4
4	1	3/64	7/32	4
3 1/2	1	3/64	1/4	3
3	1 1/8	1/16	5/16	3
2 1/2	1 5/16	1/16	11/32	3
2	1 5/8	3/32	7/16	3
1 3/4	1 7/8	1/8	1/2	3
1 1/2	2 3/16	5/32	9/16	3
1 1/4	2 9/16	3/16	11/16	3

TABLE 3 DIMENSIONS OF GROOVE

(Hob set at an angle to axis equal to helix angle minus angle of hob)

Diametral pitch, inches	A, inches	B, inches	C, inches	No. of threads in hob
10	7/8	1/16	5/32	1
8	7/8	1/16	5/32	1
7	7/8	1/16	5/32	1
6	15/16	1/16	5/32	1
5	1 1/8	1/16	3/16	1
4	1 5/16	3/32	7/32	1
3 1/2	1 7/16	3/32	1/4	1
3	1 5/8	1/8	5/16	1
2 3/4	1 13/16	1/8	21/64	1
2 1/2	2	5/32	11/32	1
2 1/4	2 5/16	5/32	3/8	1
2	2 3/8	3/16	7/16	1
1 3/4	2 11/16	3/16	1/2	1
1 1/2	2 7/8	3/16	9/16	1
1 1/4	3 1/8	1/4	11/16	1

TABLE 4 TOOTH-FORM AND SPEED FACTORS

Values of Y				Values of K: $\left(K = \dfrac{3000}{3000+V}\right)$			
No. of teeth	Y	No. of teeth	Y	V	K	V	K
12	0.078	27	0.111	0	1.000	1100	0.731
13	0.083	30	0.114	100	0.968	1200	0.714
14	0.088	34	0.118	200	0.938	1300	0.697
15	0.092	38	0.122	300	0.909	1400	0.681
16	0.094	43	0.126	400	0.882	1500	0.666
17	0.096	50	0.130	500	0.857	1600	0.652
18	0.098	60	0.134	600	0.833	1700	0.638
19	0.100	75	0.138	700	0.811	1800	0.625
20	0.102	100	0.142	800	0.789	1900	0.612
21	0.104	150	0.146	900	0.769	2000	0.600
23	0.106	300	0.150	1000	0.750		
25	0.108	Rack	0.154				

VALUES OF S

Chrome-nickel heat-treated steel, (S.A.E. 3245)	30,000
50 C forged steel (S.A.E. 1045)	25,000
30 C forged steel (S.A.E. 1030)	20,000
Cast steel, A.S.T.M. Class A	15,000
Gray iron	8,000
Bronze, 88–10–2	8,000

12 *Groove.* The width and depth of groove in center of blank for gears cut with *single-threaded* hobs set at proper angle with axis of gear are given in Table 3. The letters A, B, and C refer to Fig. 1.

13 *Groove.* The width and depth of groove in center of blank

(*Continued on page 146*)

Engineering and Industrial Standardization

Coöperation between Federal Specifications Board and Industry Promoted by A.E.S.C.

AN IMPORTANT step toward the elimination of the differences between specifications for Government purchases and the usual practice of commercial suppliers has been taken through the appointment by the American Engineering Standards Committee of a standing Committee on Coöperation with the Federal Specifications Board. Such differences in practice between Government and commercial orders are often responsible for the statement, common in commercial circles, that it costs 10 per cent more to do business with the Government than with other customers.

The members of the Committee on Coöperation with the Federal Specifications Board are:

A. H. HALL, *Chairman*, assistant treasurer and superintendent of distribution, Central Union Gas Company, New York City.
JOHN A. CAPP, chief of the testing laboratory, General Electric Company, Schenectady, New York.
SULLIVAN W. JONES, chairman, Structural Service Committee, American Institute of Architects, 19 West 44th Street, New York City.

The appointment of this committee is the culmination of conferences between Dr. S. W. Stratton, Chairman of the Federal Specifications Board, other Government officials, and representatives of industry extending over a period of several months. It is expected that in future editions a large part of the specifications will go through the regular procedure of the American Engineering Standards Committee, in order that the industrial and government specifications may be unified, resulting in truly national specifications recognized by industry and Government alike.

It is the unanimous opinion of the Federal officials and the officers of the A.E.S.C. that this coöperation between manufacturing industries, the Government and other consumers produced a widening of the source of supply for all Government requirements and in economies running into many millions of dollars yearly.

The Federal Government is one of the largest purchasers of industrial products, both as to the amount and the range of supplies. The diversity of the various Government specifications, their departures from usual commercial production, and the special features frequently required became very troublesome to the manufacturing industries; this remained true to a large extent until very recently. This Board has adopted nearly 40 specifications and is actively at work on many more.

It is the aim of the Federal Specifications Board to bring into line with the best commercial practice, not only the specifications which are to be approved by the board in the future, but also those which have already been approved. The latter will be accomplished when the approved specifications come up for review and possible revision. Such review is planned at intervals of one year.

System for Numbering Steels to be Developed Under Procedure of A.E.S.C.

A SYSTEM of designating kinds or qualities of steels by code numbers, each of which would represent a definite specification, will be developed as a result of the decision of a conference of the principal producers and users of steel held at Washington, D. C., December 6, at the call of the American Engineering Standards Committee. The conference recommended that this code be developed under the procedure of the A.E.S.C. and suggested to that organization the appointment of the Society of Automotive Engineers and the American Society for Testing Materials as joint sponsors for the code.

The agreement to go ahead with this project was arrived at after a spirited discussion concerning the necessity for and practicability of a numbering system. Strong opinions in favor of the designation of steels by number were voiced by heavy buying interests, such as the U. S. Navy Department, the Electrical Manufacturers' Council, the Society of Naval Architects and Marine Engineers, the U. S. War Department, and the Federal Specifications Board. It was pointed out during this discussion that shipbuilders use every conceivable variety of steel. Opposition to the inclusion of tool steel was voiced by tool-steel makers.

The conference voted that it is desirable to have a uniform numbering system, based on specifications, for forging steels, casting steels, structural steels including plates, tool steels, and other steels, this decision, with the exception of tool steels, being taken without dissent. Whether the basis for such a numbering system should be chemical composition, physical properties, or heat treatment was left to be determined by a Sectional Committee the personnel of which is to be approved by the American Engineering Standards Committee. It was also left to the Sectional Committee to decide whether there are any existing systems which can be used as a basis for numbering codes for any or all of the various groups of steels. The question of whether brand names can be accommodated to and associated with a numbering system was brought up, but the consensus of opinion was that this is not practicable.

The conference was opened by a résumé of present American practice in designating steels by Dr. G. K. Burgess, Chief of the Division of Metallurgy of the U. S. Bureau of Standards, and a résumé of European practice by L. H. Fry, representing the American Society for Testing Materials. Mr. Fry said that Switzerland and Germany have already taken definite steps toward a numbering system. The method proposed in Switzerland provides a system of symbols intended to be universal and definite, and capable of expansion to suit new requirements. In France a method is offered by which steels will be numbered with relation to a definite specification for the type, augmented by a letter indicating the method of manufacture, and a number showing the minimum tensile strength. In Great Britain a numbering system is employed for aircraft steels, and a tendency is appearing to develop symbols for automobile steels. Some limited symbolization is used in Holland also. A copy of Mr. Fry's full report, which is based on information obtained from abroad by the A.E.S.C., will be sent upon request to the American Engineering Standards Committee.

Dr. Burgess presided at the conference, which was attended by thirty-two men representing twenty-two organizations.

The recommendations of the conference were presented to the American Engineering Standards Committee, at its meeting on December 14. The Committee took the following action:

Resolved, That the American Society for Testing Materials and the Society of Automotive Engineers be designated joint sponsors for the development of a numbering system for forging, casting, and structural steels, including plates, but not including tool steels, the numbering system to be based on definite specifications, and—

Resolved, That for the present the question of developing such a numbering system for tool steels be held in abeyance.

Standards for Herringbone Gears

(*Continued from page 145*)

for gears cut by planing or shaping methods equal respectively:

$$\text{Width} = 1^1/_8 \text{ in.}$$

$$\text{Depth} = \frac{1.8}{P} + \text{suitable clearance.}$$

14 *Horsepower Rating.* For Pitch-Line Velocities up to 2000 Feet per Minute:

$$\text{Horsepower} = \frac{WV}{33{,}000}$$

where $W = \frac{S}{2} PFYK$

(Formula safe for wear when well oiled. For grease lubrication multiply by factor of from 0.8 to 0.62)

W = load in pounds
S = working stress (no speed) (See Table 4)
P = circular pitch
Y = tooth-form factor from Table 4
F = width of active face in inches. (For continuous contact of teeth, minimum face = 6 P)
K = speed factor from Table 4
V = pitch-line velocity in feet per minute

LIBRARY NOTES AND BOOK REVIEWS

A Dictionary of Applied Physics

Dictionary of Applied Physics, Edited by Sir Richard Glazebrook, in 5 Volumes. Vols. 1 (Mechanics, Engineering, Heat) and 2 (Electricity) Macmillan Co., London, 1922. Cloth, 6 × 9 in., 1067 pp. and 1104 pp., illus. Price $15.00 each.

A fundamental work, the various sections of which have been prepared by men of international reputation in their particular lines.

The mechanical engineer will be particularly interested in the first volume, which covers mechanics, engineering, and heat. As an illustration of the extensiveness of treatment of the various subjects considered, it may be noted that the physics of the steam turbine are treated by Gerald Stoney and Telford Petri in a 20-page article, and that the development of the steam turbine is discussed by Robert Dowson in an additional 18 pages. Other subjects are dealt with on a similar generous scale, both as regards the space devoted to them and thoroughness of handling. Many of the articles represent a successful attempt to bring together in a concise and craftsmanlike form the latest information on recent tendencies in machine design. Such, for example, are the article on the balancing of engines and prime movers (Vol. 1, pp. 252–267), by Wm. Ernest Dalby and the extremely interesting contribution on elastic constants (Vol. 1, pp. 215–241) by Reginald G. Batson.

One cannot help feeling, however, that certain subjects have been treated all too briefly. For example, the eleven pages devoted to the theory of elasticity by R. V. Southwell, notwithstanding their interest, do not present the matter as completely as the importance of the subject warrants.

It would appear generally that the Dictionary of Applied Physics places limits on the space devoted to strictly engineering developments that have been extensively covered by treatises: for example, the whole subject of refrigeration receives only 13 pages, and even the important matter of cryogenation (liquefaction of gases) is handled in eight pages; on the other hand, the subject of the realization of an absolute scale of temperature is given thirty-five pages.

Notwithstanding this apparent (and it is only apparent) inequality of treatment, the two volumes thus far issued represent not only a tremendous amount of work, but what is more, a fairly complete summary of our knowledge of applied physics within the range treated, and as such they are on a par with works such as Thompson and Tait's Natural Philosophy.

Among the subjects to which considerable space is devoted the following may be cited: Ship Resistance, Geo. S. Baker; Kinematics of Machinery, Chas. H. Bulleid; Reciprocating Steam Engine, Andrew Cruickshank; The Water-Cooled Petrol Engine, Aubrey T. Evans; Simple Harmonic Motion, Horace Lamb; Stream-Line Motion, Horace Lamb; Friction, T. E. Stanton; Thermodynamics of Internal-Combustion Engines, Sir Dugald Clerk; Theory of Steam Engine, Sir James A. Ewing; Latent Heat, Ezer Griffiths; Conduction of Heat: Mathematical Theory, Horace Lamb; Thermal Expansion, Alfred Wm. Porter; and Specific Heat of Gases, David R. Pye.

Analysis of Rubber. By John B. Tuttle. Chemical Catalog Co., N. Y., 1922. (American Chemical Society. Monograph Series.) Cloth, 6×9 in., 155 pp., $2.50.

A detailed critical summary of methods for the analysis of rubber and rubber goods, addressed primarily to chemists in consumers' laboratories and to those without previous experience in the technology or analysis of rubber. Includes an extensive bibliography. In addition to analytical methods, the book contains brief accounts of the composition of crude rubber, the preparation of rubber compounds, the theory of vulcanization and methods for physical testing.

Chauffage des Chaudières au Charbon Pulvérisé. By Michel Sohm. Chaleur et Industrie, Paris, 1922. Paper, 9×11 in., 38 pp., illus., diagrams, 4 fr. 25.

The Compagnie des Mines de Bruay has recently installed, in its electric plant at Labuissière, a powdered-coal plant for firing its boilers, which is one of the largest in Europe. This pamphlet, by the engineer in charge, describes the plant in great detail, together with its constituent parts, gives a record of the results obtained in operation and conclusions as to the merits of pulverized coal.

Diesel Engines for Land and Marine Work. By A. P. Chalkley. Fifth edition. D. Van Nostrand, N. Y., 1922. Cloth, 6×9 in., 330 pp., illus., diagrams, tables, $6.

Since the last edition of this work was published in 1915, the Diesel engine has been greatly developed and adopted much more widely. Many new manufacturers have arisen and many new types have been developed. In the present revised edition, these new engines have been dealt with as fully as possible and their salient features described. In addition to descriptive matter, the theory, installation, testing, operation and design of Diesel engines are treated.

Elasticity and Strength of Materials Used in Engineering Construction. Section 1. By C. A. P. Turner. Published by the Author, Minneapolis, Minn., 1922. Cloth, 6×9 in., 85 pp., illus. tables.

This volume is intended to demonstrate the fundamental principles upon which rational analysis may be founded and to discuss the properties of the commonly used materials from the point of view of elasticity. It presents, for the first time, a development of the relations of residual to elastic strain sufficient to account for elastic phenomena not understood heretofore and discusses the error resulting from general misunderstanding of these relations. Here is also demonstrated, for the first time, that there is but one independent modulus of elasticity for the resistance of a homogeneous isotropic solid. The relative value of the coefficients for such a body and their modification by residual strain are investigated.

Engineering Inspection. By E. A. Allcut and Chas. J. King. Van Nostrand Company, N. Y., 1922. Cloth, 7×10 in., 187 pp., illus., diagrams, tables, $5.

Much descriptive matter has already been written on the details of the various inspection methods used in different engineering works, but the object of this work is to present a convenient description of the various principles involved in the inspection of an engineering job from the raw material to the finished job. The examples given are representative ones, illustrating the different principles of inspection and measurement in common use. The book is confined to mechanical-engineering operations.

Engineering Workshop Handbook. By Ernest Pull. Fifth edition. D. Van Nostrand, N. Y., 1922. Cloth, 4×7 in., 175 pp., illus., diagrams, tables, $5.

This little pocketbook is intended for apprentices and machinists. It includes workshop mathematics, the heat treatment of metals, descriptions of the common tools and machines and of the usual operations, lathe work, screw cutting, drilling, planing, milling, grinding, etc. The book will be useful to beginners.

Etude sur le Ballon Captif et les Aéronefs Marins. By Charles Lafon. Gauthier-Villars et Cie, Paris, 1922. Paper, 7×10 in., 206 pp., 20 fr.

Although the elongated captive balloons known as "sausages" have been used by the French navy since 1917, there has not existed, until now, an exact theory of the equilibrium of the balloons watching over moving vessels. The present volume presents important scientific ideas upon towed balloons and naval aerial tactics. It also contains new data and diagrams of interest in the navigation and maneuvers of naval forces.

FLOW OF GASES IN FURNACES. By W. E. Groume-Grjimailo. With an Appendix upon The Design of Open-Hearth Furnaces, by A. D. Williams. John Wiley & Sons, N. Y., 1923. Cloth, 6×9 in., 399 pp., illus., $5.50.

Upon the basis of extended observations in practice and careful experimental researches, Professor Groume-Grjimailo has formulated a hydraulic theory of flow of heated gases, developed the laws governing this flow, and shown how these may be applied in designing furnaces for industrial purposes. Extensive appendices, amounting to over one-half of the text, contain formulas and tables upon gas flow and arch brickwork, together with several articles by Mr. Williams, treating of the design of open-hearth furnaces and hot-blast stoves, combustion and boiler settings, and heat-capacity and calorific-intensity curves.

LA FORCE MOTRICE ELECTRIQUE DANS L'INDUSTRIE. By Eugéne Marec. Gauthier-Villars et Cie., Paris, 1922. Paper, 7×10 in., 613 pp., illus., diagrams, 55 fr.

A practical treatise for engineers interested in the industrial application of electric power. Intended to explain the fundamental characteristics of commercial electric machines, in order to facilitate the choice of proper equipment, and to explain how this equipment should be installed, operated, and maintained. Includes descriptions of accessory machinery and equipment, and shows many examples of the application of motor drive to various classes of machinery.

LES HÉLICOPTÈRES. By W. Margoulis. Gauthier-Villars et Cie., Paris 1922. Paper, 6×9 in., 90 pp., diagrams, 10 fr

Part one of this volume is based on M. Margoulis's experimental investigation of screw propellers, which has enabled him to trace, for the first time, the characteristic curves of the most general action of a screw propeller. In the second part, the results of experience are applied to the study of the flight of a helicopter, in vertical and oblique flight.

INDUSTRIAL APPLICATIONS OF X-RAYS. By P. H. S. Kempton. Isaac Pitman & Sons, London, 1922. (Pitman's Technical Primers.) Cloth, 4×7 in., 112 pp., illus., 2s. 6d.

A concise account of the apparatus used to produce X-rays and the methods of using them to examine metals, non-metals and composite structures of various kinds. A bibliography is also included.

INTRODUCTION TO THE CALCULUS. By William F. Osgood. Macmillan Company, New York, 1922. Cloth, 5×8 in., 449 pp., $[illegible].90

A revision of the author's First Course in the Differential and Integral Calculus. The plan of treatment is the same, but the presentation is fuller. The objects of the book are to set forth the application of the calculus to problems of geometry and physics of the first order of importance, and to make clear the thought that underlies the calculus. The book is intended for the engineer, the physicist and the student of pure mathematics.

LUBRICATION AND LUBRICANTS. By J. H. Hyde. Isaac Pitman & Sons, London, 1922. (Pitman's Technical Primers.) Cloth, 4×7 in., 114 pp., illus., diagrams, tables, 7×4 in., cloth, $0.85.

Outlines, in as simple a manner as possible, consistent with the assumption of a general knowledge of engineering terms, the function of lubricants, the types, their application, the nature of friction and the theory of lubrication. Methods of mechanical, physical and chemical testing are described, recent developments in lubrication are discussed and a number of important examples of lubrication and lubricators are given.

MEASUREMENT OF GAS AND LIQUIDS BY ORIFICE METER. By Henry P. Westcott and John C. Diehl. Second edition. Metric Metal Works, Erie, Pa., 1922. Fabrikoid, 5×8 in , 434 pp., illus., diagrams, tables, $4.50.

Authoritative information on the orifice meter and its uses for measuring air, steam, water and oil. The first section of the book gives general information on the meter. Section two discusses the physical properties of fluids. Section three is devoted to measurement by orifice meter, while the remaining sections describe its application for various purposes.

MECHANICAL HANDLING AND STORING OF MATERIAL. By George Frederick Zimmer. Third edition. D. Van Nostrand, N. Y., 1922. Cloth, 8×11 in., 804 pp., illus., diagrams, $15.

An elaborate monograph on methods and machines for handling materials continuously or intermittently, for loading and unloading, and for automatic weighing. The most complete work in English on elevating and conveying machines, cableways, bridge cranes, warehousing machinery, etc. In this edition obsolete installations and devices have been deleted or replaced by modern examples, and the examples of American or Continental practice have been replaced by examples of British manufacture. The text has been carefully revised throughout.

MOTOR VEHICLE ENGINEERING; THE CHASSIS. By Ethelbert Favary. First edition. McGraw-Hill Book Co., New York, 1922. Cloth, 6 × 9 in., 468 pp., illus., diagrams, $5.

In this volume, as in the preceding volume on engines, the author endeavors to present, in simple language and with only elementary mathematics, the information required by the automotive engineer and the designer. It is not intended as a history of automobile development, nor as a mere description of present practice as exemplified in the vehicles now being manufactured, but to explain the theory that underlies the design of the automobile chassis.

PRODUCTION ENGINEERING AND COST KEEPING FOR MACHINE SHOPS. By William R. Basset and Johnson Heywood. First edition. McGraw-Hill Book Co., New York, 1922. Cloth, 6×9 in., 311 pp., illus., $3.50.

Written to bring practical assistance to the production managers, foremen and cost accountants of machine shops, and to give higher executives a knowledge of good shop-management practice, so that they may judge the effectiveness of the methods of their subordinates. Methods are given in detail, and the reasons explained. Language devoid of technicalities is used. The costkeeping methods described are suitable, in the author's opinion, for nine-tenths of all the American machine shops.

PRODUCTION GRINDING. By Fred B. Jacobs. Penton Publishing Co., Cleveland, 1922. Cloth 6 × 9 in., 238 pp., illus., $3.

Describes the methods used in representative plants for cutting and finishing machine parts. Includes the practice of the Marmon, Packard, Oakland, Chevrolet, and Ford automobile plants, methods for finishing chilled-iron valve cams, ball and roller bearing production, grinding operations in the making of dental tools, finishing calender rolls, reconditioning automobile engines, die grinding and the making of milling cutters. Full data are given concerning methods and output.

RAILROAD CONSTRUCTION. By Walter Loring Webb. Seventh edition, revised and enlarged. John Wiley & Sons, New York, 1922. Fabrikoid, 4 × 7 in., 847 pp., illus., diagrams, tables, $5.

Important changes and additions have been made in relation to the shrinkage of embankments; the laws governing the life of ties; substitutes for wooden ties; rail specifications, testing, failures, wear; rail-joint failures; water-tank construction; hump yards, yard grades; train resistance; and stresses in track.

REIGN OF RUBBER. By William C. Geer. Century Co., New York, 1922. Cloth, 5 × 8 in., 344 pp., illus., map, $3.

An interesting account of the rubber industry, written in popular style and fully illustrated. Describes the development of the industry from its beginnings and explains how certain representative rubber products, such as tires, footwear and clothing, golf balls, electric conductors, hose, belting, packing and balloon fabrics, are made. Intended for users of rubber goods, rather than for makers.

RESEARCHES ON CELLULOSE, Vol. 4, 1910–1921. By Charles F. Cross and Charles Dorée. Longmans, Green & Co., New York, 1922. Cloth, 5 × 8 in., 253 pp., diagram, tables, $5.

These volumes of researches, of which this is the fourth, are intended as supplements to the original volume on cellulose, published in 1895. They provide a review and critical evaluation of the research work upon the constitution and properties of cellulose which has been published in various technical journals, together with certain new matter contributed by the reviewers.

SHORT HANDBOOK OF OIL ANALYSIS. By Augustus H. Gill. Tenth edition. Lippincott, Philadelphia, 1922. Cloth, 5×8 in., 223 pp., tables, $2.50.

A concise manual, primarily for beginners, giving the methods of applying the usual physical and chemical tests to mineral, animal, and vegetable oils. Discusses only the more commonly occurring oils, as regards their preparation, properties, analytical constants, uses and adulterants. This edition has been revised and a section on motor gasoline and a description of the MacMichael absolute viscosimeter have been added.

SHIELD AND COMPRESSED-AIR TUNNELING. By B. H. M. Hewett and S. Johannesson. First edition. McGraw-Hill Book Co., New York, 1922. Cloth, 6 × 9 in., 465 pp., illus., diagrams, tables, $5.

The authors state that so much tunneling has been done with the aid of the shield and the art of such work has been developed to such a point that it is now possible to formulate, to some extent at least, certain principles and rules of practice on which to base the design and construction methods of future work. This they have tried to do in the present book, which treats of tunnel design, stresses in tunnel linings, methods of lining, tunnel shields, plant and equipment, methods of construction, maintenance and inspection, progress, cost, and surveying. A chapter is devoted to compressed-air sickness. A good bibliography is included.

SPARKING PLUGS. By A. P. Young and H. Warren. Isaac Pitman & Sons, New York, 1922. (Pitman's Technical Primer Series.) Cloth, 4 × 6 in., 106 pp., illus., diagram, tables, $0.85.

This little book reviews briefly the history of spark plugs and discusses the general principles of electric ignition. The operation and design of spark plugs is then considered in detail. The questions of electrodes and voltages are treated in detail, with data from researches by the authors. This is followed by a consideration of the composition and properties of spark-plug insulators.

STEAM TURBINE; THEORY AND PRACTICE. By William J. Kearton. Isaac Pitman, New York, 1922. Cloth, 6 × 9 in., 456 pp., illus., diagrams, $4.50.

In the opinion of Mr. Kearton, many recent books on the steam turbine have been over-developed along certain lines, and few have been generally useful to the student. The present work is written for the student, and should prove useful to engineers and draftsmen desiring a wider knowledge of theory.

After introductory chapters on the properties of steam and entropy, the book treats of the steam-turbine cycle, the flow of steam through nozzles and blades, efficiency, lubrication, stresses, critical speed, and turbine design. This is followed by descriptions of commercial turbines of various types.

STEAM-ENGINE PRINCIPLES AND PRACTICE. Terrell Croft, editor. First edition. McGraw-Hill Book Co., N. Y., 1922. Cloth, 6×8 in., 513 pp., illus., diagrams, $3.50.

This book has been very carefully prepared, the author states, to satisfy the demand for a "practical" book containing the information concerning steam engines which is wanted by an operating engineer or plant superintendent. Nothing pertaining to design has been included. The treatment has been directed to the selection, operation, care and repair of steam engines and to questions of economic operation.

A.S.M.E. Boiler Code Committee Work

(Continued from page 137)

integral part of the boiler? The thickness for heads which have riveted construction is not definitely specified in this paragraph.

Reply: It is the opinion of the Boiler Code Committee that the term "riveted shells" in the first sentence of this paragraph is intended to cover heads, in case they are not used as tube sheets.

CASE No. 409

Inquiry: Is it permissible, under Par. M-11 of the Code for Miniature Boilers, to weld on to the shell a half-coupling for the reinforcement to give the necessary four full threads?

Reply: It is the opinion of the Committee that the reinforcing may be accomplished only by a riveted pad or its equivalent, or by building up the thickness of the plate by welding, and that the welding of the half-coupling on the outside of the plate is not permissible under this requirement.

CASE No. 410

Inquiry: Is it the intent of Par. M-16 of the Code for Miniature Boilers that non-ferrous metal shall be used for the valve seat and also that the lifting device shall be of non-ferrous metal, as are required by Pars. 282 and 283 of the Power Boiler Section of the Code?

Reply: Attention is called to the fact that where the Rules of the Code for Miniature Boilers do not apply, those as above referred to in the Power Boiler Section of the Code are applicable.

Some Technical Bibliographies

THE following list of bibliographies on technical subjects has been compiled by Raymond N. Brown, of the Engineering Societies Library, 29 West 39th St., New York, N. Y., where all of the publications named are to be found. The photostatic service as described in THE ENGINEERING INDEX may also be used in connection with this index. Bibliographies upon any technical subjects will be compiled by the Library, upon request, at the rate of two dollars an hour.

Alcohol, Power. Power Alcohol, Its Production and Utilization. H. Frowde, London, 1922, 323 pp. Extensive lists of references are found at the end of each chapter.

Beams, Vierendeel System, Calculation of. Statik der Vierendeeltrager, Karl Kriso. J. Springer, Berlin, 1922, 288 pp.; bibliography, pp. 287–288. About 50 references, mostly to German publications.

Cast Steel. Der Stahlguss als Werkstoff, Rudolf Schafer. Giesserei-Zeitung, vol. 19, pp. 463–472 and 474–482; bibliography, p. 482. Sixty-four references, all in German.

Chrome Refractories. Chrome Refractories, J. S. McDowell and H. S. Robertson. American Ceramic Society, Journal, vol. 5, pp. 865–887; bibliography, pp. 882–887. About 130 references arranged chronologically.

Combustion, Surface. Surface Combustion, R. F. Bacon and W. A. Hamor. American Fuels, 2 vols., McGraw-Hill Co., Inc., New York, 1922. Bibliography in vol. 2, pp. 1091–1094. About 80 references arranged chronologically.

Duralumin. Duralumin: A Digest of Information, H. C. Knerr. American Society for Steel Treating, vol. 3, pp. 13–42; bibliography, pp. 41–42. Twenty-eight references.

Dust Explosions. Dust Explosions: Theory and Nature of Phenomena; Causes and Methods of Prevention. National Fire Protection Association, Boston, 1922, 246 pp.; bibliography, pp. 241–246. About 150 references arranged chronologically.

Graphic Charts. Graphic Charts in Business—How to Make and Use Them, A. C. Haskell and J. G. Breaznell. Codex Book Co., Inc., New York, 1922, 250 pp.; bibliography, pp. 239–246. Over 40 books and about 140 periodical articles are listed. The latter are classified.

Lumber. Lumber, Its Manufacture and Distribution, R. C. Bryant. John Wiley & Sons, Inc., 1922, 539 pp.; bibliography, pp. 439–448. About 180 references arranged by classes.

Metals, Non-Ferrous, Season Cracking of. Bibliography on Season Cracking. British Non-Ferrous Metals Research Association, Bulletin, no. 6, July, 1922, pp. 14–18. Forty-seven references in chronological order.

Partitions, Sound-Proof. Sound-Proof Partitions, F. R. Watson. University of Illinois, Engineering Experiment Station, Urbana, Ill., Bulletin 127, 1922, 85 pp.; bibliography, pp. 77–78. Thirty-four references.

Steel, High-Speed Tool. Bibliography of High-Speed Tool Steels. American Society for Steel Treating, Transactions, vol. 3, pp. 47–89. About 360 references with descriptive notes covering period from 1900 to May, 1922. Divided into parts entitled: I, Manufacture of High-Speed Steel; II, Heat Treatment; Uses and Tests. Arranged chronologically in each part.

Steel, Rustless. Rostfreie Stahle, Karl Daeves. Stahl und Eisen, vol. 42, pp. 1315–1320; bibliography, pp. 1319–1320. Thirty-two references.

THE ENGINEERING INDEX

(Registered United States, Great Britain and Canada)

Exigencies of publication make it necessary to put the main body of The Engineering Index (p. 101-EI of the advertising section) into type considerably in advance of the date of issue of "Mechanical Engineering." To bring this service more nearly up to date is the purpose of this supplementary page of items covering the more important articles appearing in journals received up to the third day prior to going to press.

AERODYNAMICS

Standardization. Standardization and Aerodynamics, William Knight. Aerial Age, vol. 15, no. 21, Dec. 1922, pp. 593–598. Writer refers to suggestions contained in his report to Nat. Advisory Committee for Aeronautics made in 1919 on desirability of calling a congress of representatives of leading aerodynamic laboratories, without discrimination between former allies and enemies and gives comments of the various countries on proposed congress.

AIRSHIPS

Girders. Improvements in Built-Up Airship Girders, S. H. Phillips. Aviation, vol. 13, no. 26, Dec. 25, 1922, pp. 828–830, 3 figs. With special reference to strength of girders as a whole.

AIR COMPRESSORS

Rotary. Rotary Air Compressors. Engineering, vol. 114, no. 2972, Dec. 15, 1922, pp. 740–742, 15 figs., partly on p. 744. New compressors constructed by Swiss Locomotive & Machine Works, Winterthur, Switzerland.

AMMONIA CONDENSERS

Comparative Reports. Comparative Ammonia Condenser Reports Point Out Inefficiencies, P. Wilson Evans. Power, vol. 57, no. 1, Jan. 2, 1923, pp. 28–30, 1 fig. By compiling weekly averages of ammonia condenser temperatures and pressures from various plants operated by Armour & Co., it has been possible to detect and remedy inefficient operating conditions with resultant saving in power and operating expense.

BLAST FURNACES

Air Intake, Elevated. Elevated Intake for Blast Furnace Air, Alfred Gradenwitz. Iron Age, vol. 110, no. 26, Dec. 28, 1922, p. 1700, 1 fig. Tests at Rombach Iron Works, Coblentz, Germany, with view to ascertaining whether upper strata of air are cooler and less capable of absorbing water than those in immediate vicinity of ground; 138-ft. suction tower for air intake affords considerable reduction of moisture.

BOILER ROOMS

Efficiency in. Modern Management Methods in the Power Plant, Browning Robinson. Indus. Management (N. Y.), vol. 65, no. 1, Jan. 1923, pp. 57–61, 2 figs. Problems of boiler-room efficiency.

BOILERS

Design. Principles of Boiler Design, C. E. Stromeyer. Engineer, vol. 134, no. 3495, Dec. 22, 1922, pp. 672–673, 1 fig. High-furnace-temperature effects; convection and velocity; costly natural draft; air heaters; superheaters; economizers; high velocity of flames; reductions of furnace temperatures; oil burning.

BONUS SYSTEMS

Packard Motor Co. A New Departmental Bonus System, E. F. Roberts. Indus. Management (N. Y.), vol. 65, no. 1, Jan. 1923, pp. 43–45, 3 figs. Method adopted by Packard Motor Car Co. for stimulating efficiency.

BOXES

Cardboard, Machinery for Making. Cardboard Box-Making Machinery. Engineering, vol. 114, nos. 2965, 2966, 2967, 2968, 2969, 2970, 2971 and 2973, Oct. 27, Nov. 3, 10, 17, 24, Dec. 1, 8 and 22, 1922, pp. 517–520, 542–543, 578–579, 609–610, 640–641, 670–671, 698–699 and 760–762, 103 figs. Complete range of machinery used in manufacture of cardboard boxes.

CENTRAL STATIONS

Pulverized-Coal Burning. Cahokia Station. Power, vol. 57, no. 1, Jan. 2, 1923, pp. 20–22, 4 figs. Outstanding engineering features of new station of Union Elec. Light & Power Co., St. Louis, Mo., as example of large central station using pulverized coal as fuel.

CORROSION

Metals, Industrial. Corrosion as Affecting the Metals Used in the Mechanical Arts, W. H. Hatfield. Engineer, vol. 134, no. 3494, Dec. 15, 1922, pp. 639–643, 21 figs. Results and other data of set of experiments undertaken to decide upon relative resistance to various corroding media of typical industrial metals. Paper read before Sheffield Assn. Metallurgists & Met. Chemists.

COST ACCOUNTING

Factory. A New System of German Factory Accounting, Schar Schlesinger and Wallichs. Management Eng., vol. 4, no. 1, Jan. 1923, pp. 27–31. Describes system of cost accounting developed by Ernst Just and Elizabeth Vöhl, based on principle of applying analytical methods as substitute for experience and intuition. Presents and discusses seven propositions upon which system is based. If generally adopted a radical change will take place in industrial organizations.

Measuring Management by. Measuring Management by Cost Accounting, William R. Basset. Am. Mach., vol. 58, no. 1, Jan. 4, 1923, pp. 37–38. Effect of human factor; lack of unit of measurement; cost-finding methods considered most effective present measure.

DIE CASTINGS

Dies for. Dies for Die-castings, A. G. Carman. Machy. (N. Y.), vol. 29, no. 5, Jan. 1923, pp. 369–370, 3 figs. Specific examples of design and construction of dies for making die castings of more intricate type.

DIES

Shells, Blank Diameters of. Blank Diameters for Drawn Shells, W. L. Tryon. Machy. (N. Y.), vol. 29, no. 5, Jan. 1923, pp. 349–354, 19 figs. Formulas and graphical methods for finding blank area and diameter of various types of shells.

DIESEL ENGINES

Nobel. The Two-Stroke-Cycle Diesel's Superiority for Large Powers, Edwin Lundgren. Power, vol. 56, no. 26, Dec. 26, 1922, pp. 1015–1016, 2 figs. Points of superiority of two-stroke-cycle engine illustrated with data on Nobel-Sweden 1600-hp. Diesel.

ELECTRIC FURNACES

Levoz. Designs New Electric Furnace, R. Sylvany. Foundry, vol. 50, no. 23, Dec. 1, 1922, pp. 962–963. Principles of construction of furnaces of latest French development, by T. Levoz, employed during war in production of high-speed steel; metal melted in crucible-shaped vessel with only one opening; electrode through roof sheathed to prevent ingress of air.

ELECTRIC MOTORS

Selection and Control. Selection and Control of Motors, Roger B. Stevens. Elec. World, vol. 80, no. 27, Dec. 30, 1922, pp. 1433–1437, 7 figs. Avoidance of service interruptions without trained and expensive maintenance crew and under necessity of operating with common labor was chief problem in Am. Sugar Refining Co.'s plant.

EMPLOYMENT MANAGEMENT

Engineering Department. The Successful Operation of an Engineering Department, W. E. Irish. Indus. Management (N. Y.), vol. 65, no. 1, Jan. 1923, pp. 19–22. Plan of action in dealing with labor. Assembling and operating the "man-machine."

FLOW OF FLUIDS

Electric Measuring Instrument. Electrical Instrument Applied to Measurement of Fluids, E. H. Freeman. Power, vol. 56, no. 26, Dec. 26, 1922, pp. 1008–1009, 2 figs. Electrical integrating instrument developed by author which is practically free from voltage error; adopted by Republic Flow Meter Co., Chicago.

FORGINGS

Nickel-Chrome. Studies Nickel Chrome Forgings, T. Henry Turner. Iron Trade Rev., vol. 71, no. 25, Dec. 21, 1922, pp. 1695–1700, 7 figs. Defects found on machined surfaces of hollow cylindrical bells are traced to ingot corner segregation; investigation showed depth of these segregations extended from 2 to 3 in.: recommendations.

FOUNDATIONS

Power Stations. Foundation and Framing Design of Colfax Power Station, M. E. Thomas. Eng. News-Rec., vol. 89, no. 26, Dec. 28, 1922, pp. 1102–1105, 7 figs. Concrete mat foundation on gravel; column loads distributed by reinforced-concrete walls; special retaining wall and sea wall held by ties.

FOUNDRIES

Ford's River Rouge. Ford Principles and Practice at River Rouge, John H. Van Deventer. Indus. Management (N. Y.), vol. 65, no. 1, Jan. 1923, pp. 11, 23 figs. Plant facilities of world's largest foundry.

GEAR CUTTING

Formed Cutters. Cutting Bevel Gears with Formed Cutters, Franklin D. Jones. Machy. (N. Y.), vol. 29, no. 5, Jan. 1923, pp. 363–365, 6 figs. Brown & Sharp automatic gear-cutting machines of formed-cutter type designed for cutting either spur or bevel gearing.

GRINDING

Automobile Parts. Grinding Data from the Pierce-Arrow Shop, Fred H. Colvin. Am. Mach., vol. 57, no. 26, Dec. 28, 1922, pp. 997–998, 3 figs. Testing cylinder gages; grinding bronze connecting-rod bushings; forming steering balls; grinding aluminum transmission covers.

HYDROELECTRIC DEVELOPMENTS

Western States. Outstanding Features of Western Water-Power Development in 1922. Eng. News-Rec., vol. 90, no. 1, Jan. 4, 1923, pp. 14–16, 2 figs. Forward steps in design and construction; stabilized tendency toward municipal ownership or control; economic aspects of Western water-power situation.

INDUSTRIAL MANAGEMENT

Factory Planning. Factory Planning, H. E. Taylor. Engineering, vol. 114, nos. 2972 and 2973, Dec. 15 and 22, 1922, pp. 756–758 and 786–788, 8 figs. Survey of systems of shop planning for various methods of production. See also Eng. Production, vol. 5, no. 115, Dec. 14, 1922, pp. 557–563, 9 figs.

Machine Jobbing Shop. Management of the Machine Jobbing Shop, H. L. Wheeler. Am. Mach., vol. 58, no. 1, Jan. 4, 1923, pp. 13–18, 1 fig. Distinction between manufacturing and jobbing; systems, operations and methods; how orders are received and handled; what jobbing-shop management should avoid.

Process Planning. Notes on Process Planning. Engineering, vol. 114, no. 2973, Dec. 22, 1922, pp. 778–780, 4 figs. Principal aim when planning sequence of processes is to secure maximum economy of production as a whole.

Production Executives. What the Production Executives Should Know from the Financial Records, William M. Lyb and. Management Eng., vol. 4, no. 1, Jan. 1923, pp. 19–23. Coördination of divisional planning and operation; information for foremen; facts in regard to department operation; overhead expenses; spoilage, waste and idleness; reports in shop language; information for production executives; cost of idle equipment and time; operation of inventory; etc.

Quality Control. Quality Products, G. S. Radford. Management Eng., vol. 3, nos. 5, and 6, Nov. and Dec. 1922, pp. 261–266, 3 figs. and vol. 4, no. 1, Jan. 1923, pp. 51–56, 5 figs. Control methods used by leading manufacturers to maintain standards. Answers to questionnaire sent to number of representative firms.

LOCOMOTIVES

Developments 1922. A Year of Innovations in Locomotive Design, A. F. Steubing. Ry. Age, vol. no. 1, Jan. 6, 1923, pp. 41–43, 4 figs. Steam-turbines locomotives in Europe; mechanical draft receiving increased attention.

MACHINERY

Power, Erecting. Erecting Power Machinery Having Cast-Iron Bedplates, N. L. Rea. Power, vol. 56, no. 26, Dec. 26, 1922, pp. 1021–1023, 6 figs. Proper supporting of machine base; preparing top of foundation; handling bedplate; leveling and aligning; grouting bedplate to foundation; assembling machine and putting it in operation.

OIL FUELS

Calorific Value. The Determination of the Calorific Value of Liquid Fuels, H. Moss and W. J. Stern. Engineering, vol. 114, no. 2972, Dec. 15, 1922, pp. 729–731, 6 figs. Experiments undertaken to investigate methods of determining calorific values, with view to evolving one by means of which absolute values for any fuel undergoing engine test could be obtained quickly and accurately.

OPEN-HEARTH FURNACES

Loftus. New Type of Open-Hearth Furnace. Iron Age, vol. 110, no. 26, Dec. 28, 1922, pp. 1677–1679, 5 figs. Loftus furnace involves new way to apply blow-torch principle; details and results of unit operating several months.

PIPE

Stresses in. The Stresses in End-Supported Filled Pipes (Ueber die Spannungen in freitragenden gefüllten Rohren), E. Schwerin. Zeit. für angewandte Mathematik u. Mechanik, vol. 2, no. 5, Oct. 1922, pp. 340–353, 3 figs. Study of influence of bending and torsional strength of pipe wall, not only under stress of liquid, but also under natural weight of pipe itself.

PUMPING ENGINES

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RAILWAY MOTOR CARS

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RAILWAY SHOPS

Shortcomings. What's Wrong with the Railroad Shops? Am. Mach., vol. 57, nos. 17, 20, 23 and 25, Oct. 26, Nov. 16, Dec. 7 and 21, 1922, pp. 677–680, 755–757, 879–881 and 955–957 and vol. 58, no. 1, Jan. 4, 1923, pp. 1–3, 2 figs. Information based on critical survey of situation. Mechanical departments said to be badly handicapped by poor equipment; lack of contact with other shops. Nov. 16: Shop equipment as it is, with suggestions for more efficient use of what is available and replacing obsolete units with new ones. Dec. 7: Small-tool situation; lack of coöperation between mechanical and purchasing departments; inadequacy of tool-sharpening equipment. Dec. 21: Suggestions regarding interchange of information and standardization of parts and methods; cost-keeping systems. Jan. 4: Underlying causes of present shop troubles; how to remedy shortcomings of methods, equipment and personnel.